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SPACE STATION CONNECTION AND SEAL STUDY

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by:

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PURPOSE

This report, submitted by Environmental Research Associates in response to Contract NAS1-2164 issued by National Aeronautics and Space Administration, Langley Research Center, combines the study, evaluation and recommendations for reliable easily replaceable seals and connection components proposed for use in self-erecting, manned space stations.

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INTRODUCTION AND STUDY METHODOLOGY

The postulation of a self-erecting, manned space laboratory, and in general, the operation of any long-term manned space experiment, requires the design of reliable, easily replaceable seal and connection components. The complexity of these components grows in direct proportion to the enclosed volume of the station manned component when considered in light of the presently planned booster capability. Connections and seals are required to connect the various living and experimental modules, to provide for safe personnel transfer between enclosed portions of the station, to provide access for resupply and crew rotation and to carry utilities and information between the various station sections.

The design of these connection-seal components must consider proven, demonstrable mechanisms and seals in order to be available for the early launch date anticipated. Therefore the following study has concentrated upon illuminating those available, acceptable concepts which will fulfill this dictum. Much useful information has been gained in discussion with the various government agencies involved in space design: Wright Air Development Center; the various National Aeronautics and Space Administration Facilities,

Langley Research Center, George C. Marshall SFC; and the various commercial concerns The Marquardt Corporation, North American Aviation, Martin-Marietta Corporation, Hi-Pressure Engineering, Wiggins Co., Parker-Hannifin, The Astro Corporation; and by careful survey of the classified and open United States and foreign literature.

The general design criteria centers about providing maintainable seals and connections with the greatest reliability. This has been accomplished by modifying the base point design concept wherever required to accommodate good seal-connection practice.

The connection and seal configurations considered are of the following major types.

1. Docking and Initial Access
2. Inter-section Bulkhead Doors
3. Section Hinge Joints
4. Inter-module Locks
5. Sliding Spoke Joints
6. Rotating Spoke Joints
7. Electrical and Fluid Transfer

Connections and Seals

The initial phase of the study comprised the evaluation

of the station operation from the connection-seal standpoint and the detailing of a set of design criteria derived from the North American Aviation Corporation report SID 1-6158.

Having obtained this data, an analytical evaluation of elastomeric and metallic seals was undertaken. The relationships between seal material, diameter, deflection, permeability and interface leakage was obtained. The various attendant environmental factors were used to modify the results obtained to yield generalized numerical results for subsequent use in determining flange loadings and overall leak rates.

From this, the several base point connection and seal configurations were analyzed to yield:

1. Operating Criteria
2. Stress Values of the Seal
3. Stress Values of the Connection
4. Connection Locking Requirements

The results of the foregoing were used to modify the seal-connection application criteria and representative resultant design configurations were executed to define final criteria and operational parameters for connection and seals to be used in conjunction with the self-erecting space laboratory concept.

BASIC OPERATION OF THE STATION AND ASSUMPTIONS

The self-deploying station concept requires the automatic make-up of a number of joint-seal configurations. The station is comprised of six peripheral, cylindrical living-working modules connected to a central hub docking-zero gravity-laboratory section by three extensible radial arms. This constitutes ten areas which are separated by 'locks'. In general, the locks are composed of two inward opening pressure assisted doors and an intervening access passage completed by an (automatic or manual) joint-seal.

The base configuration as detailed by North American Aviation Corporation in SID 62-658-1 engenders the following descriptions and assumptions.

1. The station is launched in a folded configuration accompanied by a modified version of the Apollo Command module carrying a three man crew.
2. The station is automatically deployed by mechanical screw-jack actuators located at the hinges of each of the six modules.
3. The extensible spokes are free hinges at their termini and follow the motions of the modules, being locked in place upon the completion of the deployment phase.

4. The crew member in the Apollo module will perform a deployment monitoring function only and will not enter the station until the completion of deployment.
5. After booster-station separation, the Apollo module separates from the station (which station is in a folded configuration), turns end for end and engages the personnel airlocks at the central docking port.
6. The modules and the hub are pressurized to 14.7 psia before launch and remain sealed during launch.
7. The modules and hub contain sufficient air to pressurize the spokes, the resultant mean station pressure being 10 psia.
8. Airlocks have been included between each module and at the spoke end, dividing the station in ten univocal compartments.
9. The Apollo command module has been chosen as the replenishment vehicle because it contains an airlock passage, and it is the only reentry vehicle planned for operation by 1966.
10. The airlocks normally remain open except in emergency.
11. The mechanization of the joints compensates for accumulated looseness in the hinges.

12. At the completion of the deployment maneuver finder latches at each hinge joint are extended, to capture pins on the adjoining modules and are drawn together, to lock the joints and provide the seal seating force.
13. The spoke lengths are not precisely controlled during extension. Final adjustment of the spoke length is an erection crew function.
14. Preliminary design estimates by NAA state that well designed 'O' ring seals, backed by a metal bellows, or sections, of flexible materials would provide an adequate seal at the joints between various sections of the station.
15. The telescoping spokes employ an 'O' ring seal that engages as the spoke reaches its maximum travel. If leakage becomes excessive this joint can be made up with flexible tapes supplied with a non-outgassing, elastomeric additive.
16. No in-space crew assembly is required or desired.
17. No crew assistance is required in the deployment phase.
18. The station will rotate continuously at 3 r.p.m. yielding 0.2g at the periphery.
19. Docking facilities for seven Apollo modules are provided allowing a full station complement of twenty-one men.

20. The station command center is located in one module.
21. The modules and the hub are provided with individual environmental control systems.
22. The crew missions is on a 6 week recycle basis.
23. An automatic airlock is provided at each docking port.
24. During personnel entry, a pressure balance between the station and Apollo module is necessary, using bleed air valves in the airlock doors.
25. Visual monitoring capability for the experiments and the docking procedure is provided.
26. The docking port airlocks are retracted 16.0 inches internal to the station to allow turret despin required for subsequent docking.
27. The vehicle adapter is gimbal mounted at the transfer boom end, and is restrained by hydraulic locking latches entering from each side of the stowing area fixed docking points.
28. The entry passage from the Apollo is a nominal 28 inch I.D.
29. The docking procedure is as follows:
 - a. The turret is mechanically driven in a direction opposite to the space station rotation, so that it becomes essentially nonrotating with reference to inertial space.

- b. An electric drive mechanism brings the stowing boom and the attachment ring over the axis of rotation.
- c. The incoming Apollo module engages the attachment ring and if desired the attachment can be made to the central airlocks in the hub.
- d. After personnel entrance, the transfer boom moves the Apollo module over to the circumferentially located stowage position on the hub.
- e. Power to the mechanical drive is shut off and bearing friction causes the matching of the rotation of the turret and the station.
- f. The turret is indexed to stop at a predetermined position.
- g. When not in use, the booms and attachment rings are stowed in recesses around the turret.
- h. The station airlock passage is presented automatically, assisted by the internal station pressure.
- i. Automatic retraction is provided by auxiliary hydraulic actuators provided with end of travel locks.

30. Normal departure entails:
 - a. Counterrotating the turret.
 - b. Moving the departure vehicle to the central docking port.
 - c. Apollo module departure.
 - d. Rearrangement of the remaining vehicles for balance.
 - e. Turret spin-up.
31. Access to the module internal wall must be available for meteorite puncture repairs.
32. A self-sealing wall construction was investigated. The concept tested employed alternate layers of nonreacted resins and catalysts which automatically mix upon meteorite penetration. This concept could be employed to seal punctures up to 1/16 inch diameter.
33. Two complete station repressurizations of air are provided to compensate for the highly probable loss of air from the sealed sections during launch.
34. Additional cryogenic nitrogen and oxygen are provided for leakage make-up.
35. The total surface area of the space station is approximately 24,000 Ft.².
36. The expected orbital life is 2 years.

37. The allowable leak rate is 10% of the station volume per year (Verbal from NASA -LRC); SID 615-68-1 says 10% in 6 weeks.
38. The estimated leak rate for the station is 18#/day.
(NAA)
39. Calculated module temperature limits are:
 Without Spin: (-40 degrees F to +250 degrees F)
 With Spin: (-45 degrees F to +100 degrees F)
40. The calculated maximum temperature during launch is +350 degrees F for 5 minutes.
41. The operating internal wall, joint temperature is maintained at +70 degrees ⁺-2 degrees F by the enviromental control system.
42. Lock sizing dictates a maximum volume and weight limitation per package brought by the Apollo module of 2 Ft³ and 100# earth weight.

DESCRIPTION AND NUMERICAL ANALYSIS OF BASE POINT DESIGN CONFIGURATION

This section covers the analysis of the connection seal configuration proposed in the reference base point design. Particularly, the following configurations are covered:

1. DOCKING ACCESS AIRLOCK
2. SPOKE-TO-HUB INTERFACE
3. SPOKE-TO-MODULE INTERFACE
4. INTER-MODULE AIRLOCK

An operational description of each major connection is presented and each concomitant seal configuration is segregated and detailed to yield sizing and stress parameters.

1. DOCKING ACCESS AIRLOCK (Figure 3-2)

a. Description

The docking access airlock comprises an extendable, cylindrical passageway utilizing three sliding seals. The airlock portion incorporates a circular access door with a single face compression seal. The external terminus of the passageway carries an 'O' ring sleeve type seal which mates to the Apollo vehicle at the end of the axial travel.

b. Service

Medium: Air/Vacuum
Operating Pressure: 10 psia/ 10^{-11} Torr
Maximum Pressure: 20 psia/ 10^{-11} Torr
Temperature Limits: +325°F to -60°F

c. Seal Configurations

1) Sliding Seals - Passageway

Quantity: 3 per Assembly, 21 Total
Travel: 16 inch Axial Stroke
Type: 'O' Ring Elastomeric
Service Life: 7 cycles/6 weeks Min.
2 Year Max.
Seal Parameters: 32.5 inch Internal Diameter
.250 inch Section Diameter
Shore A 60 Durometer
10% Radial Deflection
Seal Reactions: 125 psi Seating Stress
867# Seating Load (Total)
1214# Static Friction Load
408# Running Friction Load

2) Sliding Seals - Apollo Mating (Diametral Deflection)

(Reference b Figure 3-2)

Quantity: 1 per Assembly, 7 Total
Travel: 4 inch linear

Service Life: Same as Passageway
Type: Same as Passageway
Seal Parameters: 30.5 inch Internal Diameter
.500 inch Section Diameter
Shore A 60 Durometer
10% Radial Deflection
Seal Reactions: 125 psi Seating Stress
1632# Seating Load (Total)
767# Static Friction Load

Actuator Requirements For Passageway and Apollo Mating

Mode: Hydraulic - Linear - End
of Travel Locks -
Speed Limited

Maximum Output Force: 14,223#

3) Face Compression Seal - Access Doors

(Reference c Figure 3-2)

Quantity: 1 per Assembly, 7 Total
Type: 'O' Ring Elastomeric
Seal Parameters: 32.5 inch Mean Diameter
.250 inch Section Diameter
Shore A 60 Durometer
20% Axial Deflection
Seal Reactions: 125 psi Seating Stress
867# Seating Load (Total)

Flange Reactions: 9163# Maximum
(case of internal hub
depressurization with a
pressurized Apollo attached)

Seating Load: Applied by friction wedges -
Hand wheel operated

2. SPOKE-TO-HUB INTERFACE (Figure 3-4)

a. Description

The spoke-to-hub interface joint seal comprises a single hinge - clamp coupled joint, sealed by a bellows loaded 'O' ring seal. The bellows acts simply as a spring to provide seal seating stress.

b. Service

Medium: Air/Vacuum

Operating Pressure: 10 psia/ 10^{-11} Torr

Maximum Pressure: 20 psia/ 10^{-11} Torr

Temperature Limits: +325°F to -60°F

c. Seal Configurations

Quantity: 1 per Assembly, 3 Total

Travel: Hinged

Type: 'O' Ring Elastomeric

Service Life: Make-up at erection for
life of station

Seal Parameters:	54.0 inch Mean Diameter
	.250 inch Section Diameter
	Shore A 60 Durometer
	20% Axial Deflection
Seal Reactions:	125 psi Seating Stress
	1445# Seating Load (Total)
	1445# Bellows Load at
	Operating Length (Appendix IV
	contains bellows design data)

3. SPOKE-TO-MODULE INTERFACE (Figure 3-5)

a. Description

The spoke-to-module interface comprises a hinge joint similar to Item 2, connecting on one end the spoke, and on the other end the spoke stub. The stub mates with the upper portions of the vestibule. As a consequence of the erection procedure, this stub section must rotate 90° relative to the module axis. This necessitates the incorporation of a rotating seal at the stub module interface.

b. Service

Medium:	Air/Vacuum
Operating Pressure:	10 psia/ 10^{-11} Torr
Maximum Pressure:	20 psia/ 10^{-11} Torr
Temperature Limits:	+325°F to -60°F

c. Seal Configurations

1) Face Compression (Reference a Figure 3-5)

Quantity: 1 per Assembly, 3 Total

Travel: None

Type: 'O' Ring Elastomeric

Service Life: Make-up at erection for
life of station

Seal Parameters: 50 inch Mean Diameter
.250 inch Section Diameter

Shore A 60 Durometer

20% Axial Deflection

Seal Reactions: 125 psi Seating Stress
1335# Seating Load (Total)
1335# Bellows Load at
Operating Length (Appendix IV
contains bellows design data)

2) Access Door (Reference b Figure 3-5)

Quantity: 1 per Assembly, 3 Total

Travel: None

Type: 'O' Ring Elastomeric

Service Life: Make-up at erection for
life of station

Seal Parameters: 40 inch Mean Diameter
.250 inch Section Diameter

	Shore A 60 Durometer
	20% Axial Deflection
Seal Reactions:	125 psi Seating Stress
	1068# Seating Load (Total)
Flange Reactions:	2325# Maximum
	(case of module depressur- ization with spoke pressur- ized)
3) Stub-Module Seal	(Reference <u>c</u> Figure 3-5
Quantity:	1 per Assembly, 3 Total
Travel:	90° Rotation during erection- Static after erection
Type:	'O' Ring Elastomeric
Service Life:	Make-up at erection for life of station
Seal Parameters:	56 inch Internal Diameter
	.250 inch Section Diameter
	Shore A 60 Durometer
	20% Axial Deflection
Seal Reactions:	125 psi Seating Stress
	1495# Seating Load (Total)
	2093# Static Friction Load
	703# Running Friction Load

4. INTER-MODULE AIRLOCK (Figure 3-6)

a. Description

The inter-module airlock forms a safe passageway between adjacent modules. One half of the lock is attached to each module end and is brought into conjunction during the final portion of the erection maneuver. Automatic interface sealing is provided by the bellows-loaded seal component. An inward opening entrance hatch is provided at the airlock module-interface and employs a face compression seal loaded by a hinge-latch.

b. Service

Medium:	Air/Vacuum
Operating Pressure:	10 psia/ 10^{-11} Torr
Maximum Pressure:	20 psia/ 10^{-11} Torr
Temperature Limits:	+325°F to -60°F

c. Seal Configurations

1) Face Compression (Reference a Figure 3-6)

Quantity	1 per Assembly, 6 Total
Travel:	None
Type:	'O' Ring Elastomeric
Service Life:	Make-up at erection for life of station
Seal Parameters:	72 inch Mean Diameter

.250 inch Section Diameter
Shore A 60 Durometer
20% Axial Deflection

Seal Reactions: 125 psi Seating Stress
1922# Seating Load (Total)
1922# Bellows Load at
Operating Length (Appendix IV
contains bellows design data)

2) Module End Access Door (Reference b Figure 3-6)

Quantity: 2 per Assembly, 12 Total

Travel: None

Type: 'O' Ring Elastomeric

Service Life: Make-up at erection for
life of station

Seal Parameters: 60 inch Mean Diameter
.250 inch Section Diameter
Shore A 60 Durometer
20% Axial Deflection

Seal Reactions: 125 psi Seating Stress
1602# Seating Load (Total)
2243# Static Friction Load
753# Running Friction Load

The preceding analysis yields a quantitative picture of the base point design configurations. It should be noted at this point that the design requires large actuators to aid in docking passageway deployment due to the combination of seal friction and pressure loading. Further, the utilization of bellows pressure loaded seals requires that the bellows provide the seal seating force of approximately 1500#. This requires the use of auxiliary actuators to compress the bellows to allow for seal replacement.

C-11

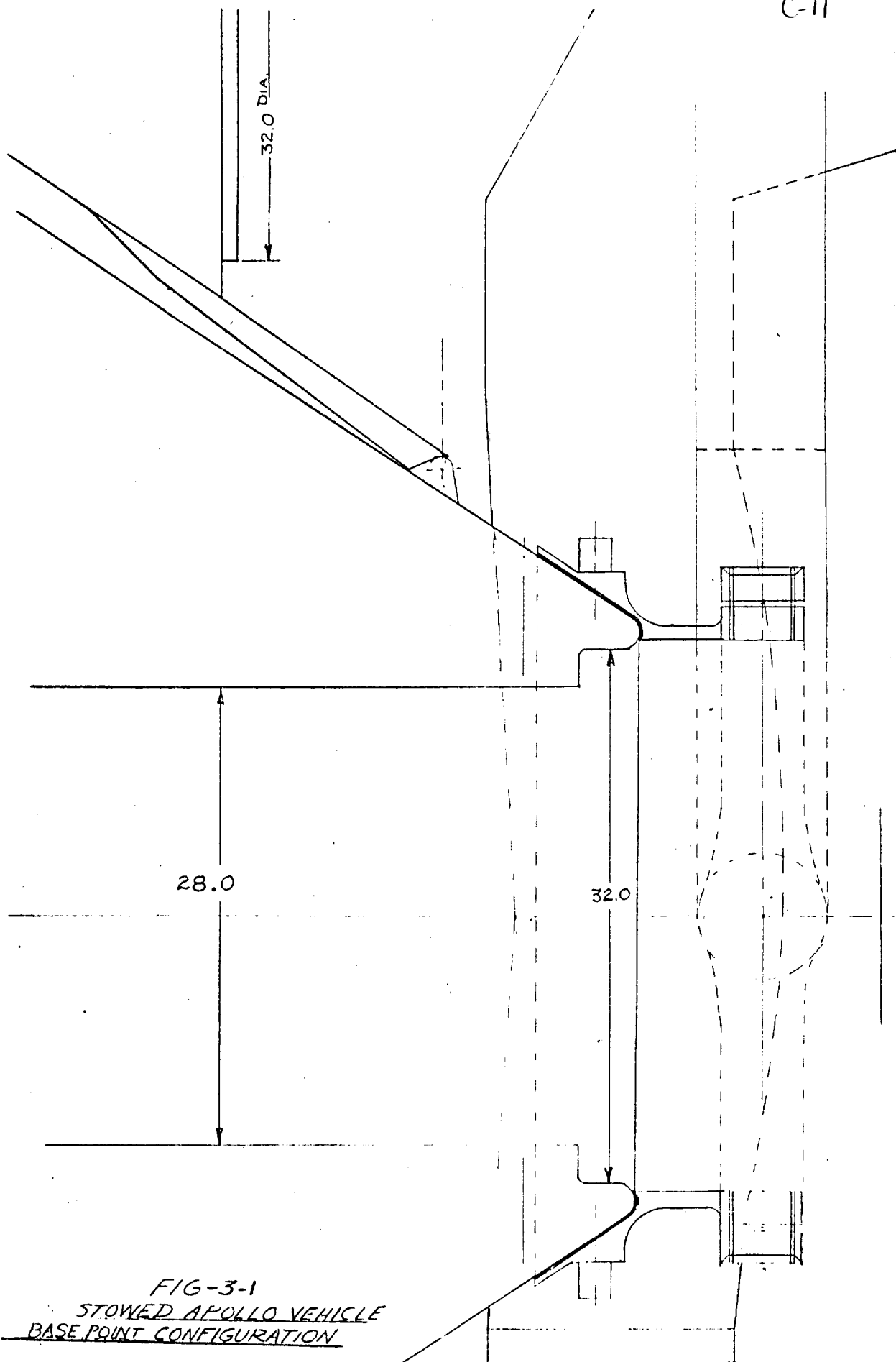


FIG-3-1
STOWED APOLLO VEHICLE
BASE POINT CONFIGURATION

C-12

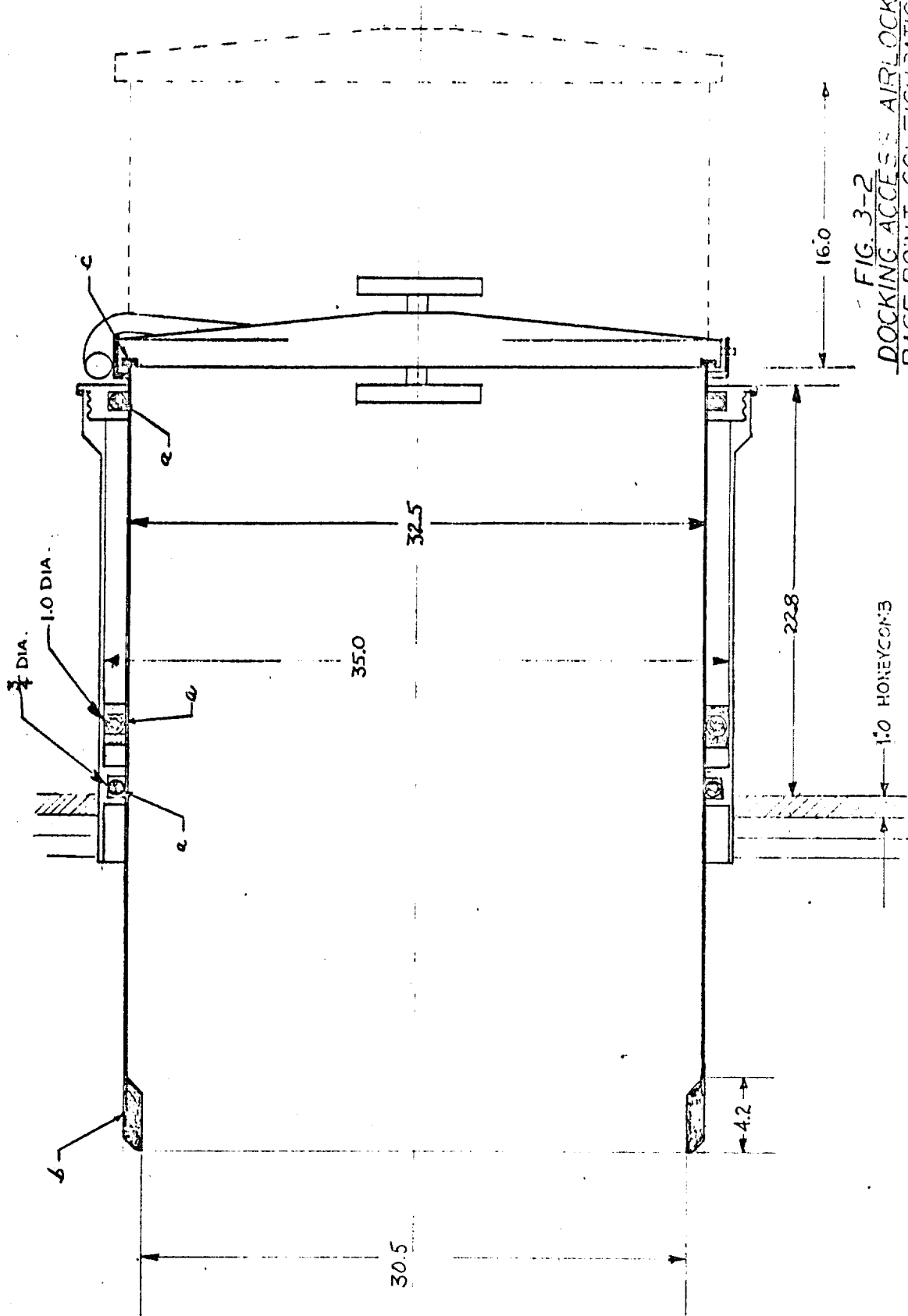
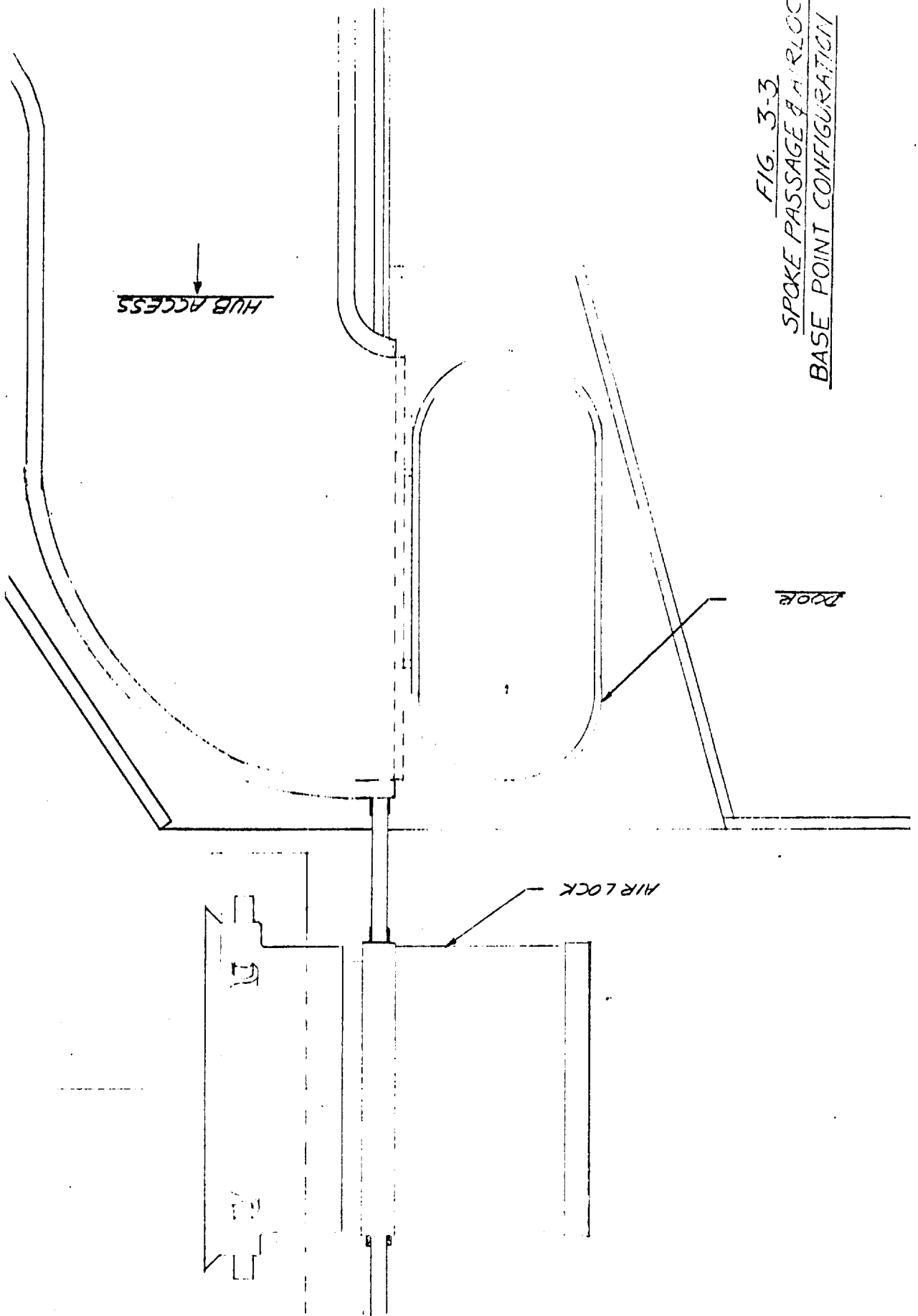
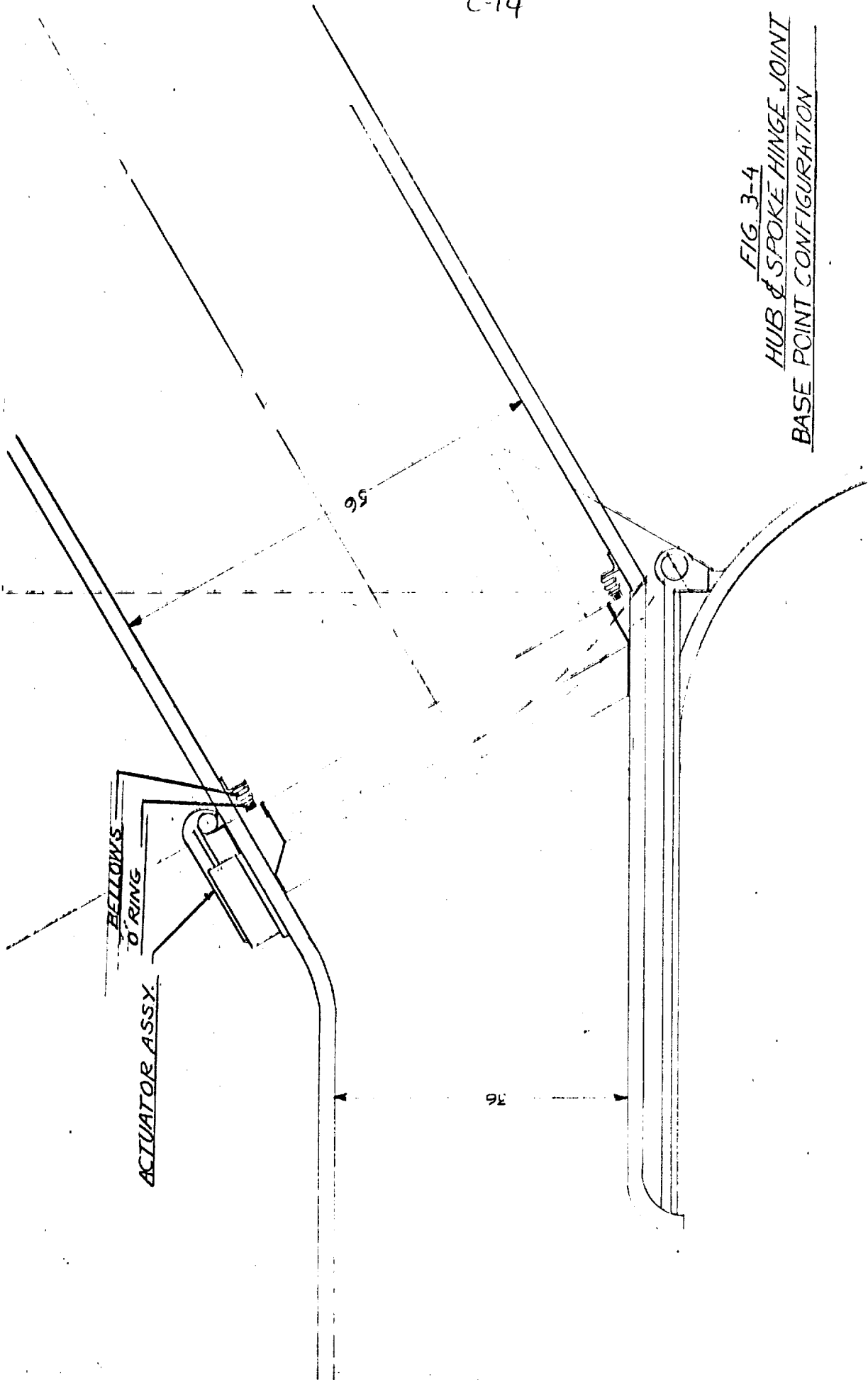


FIG. 3-3.
SPOKE PASSAGE & AIR LOCK
BASE POINT CONFIGURATION



C-14

FIG. 3-4
HUB & SPOKE HINGE JOINT
BASE POINT CONFIGURATION



C-15

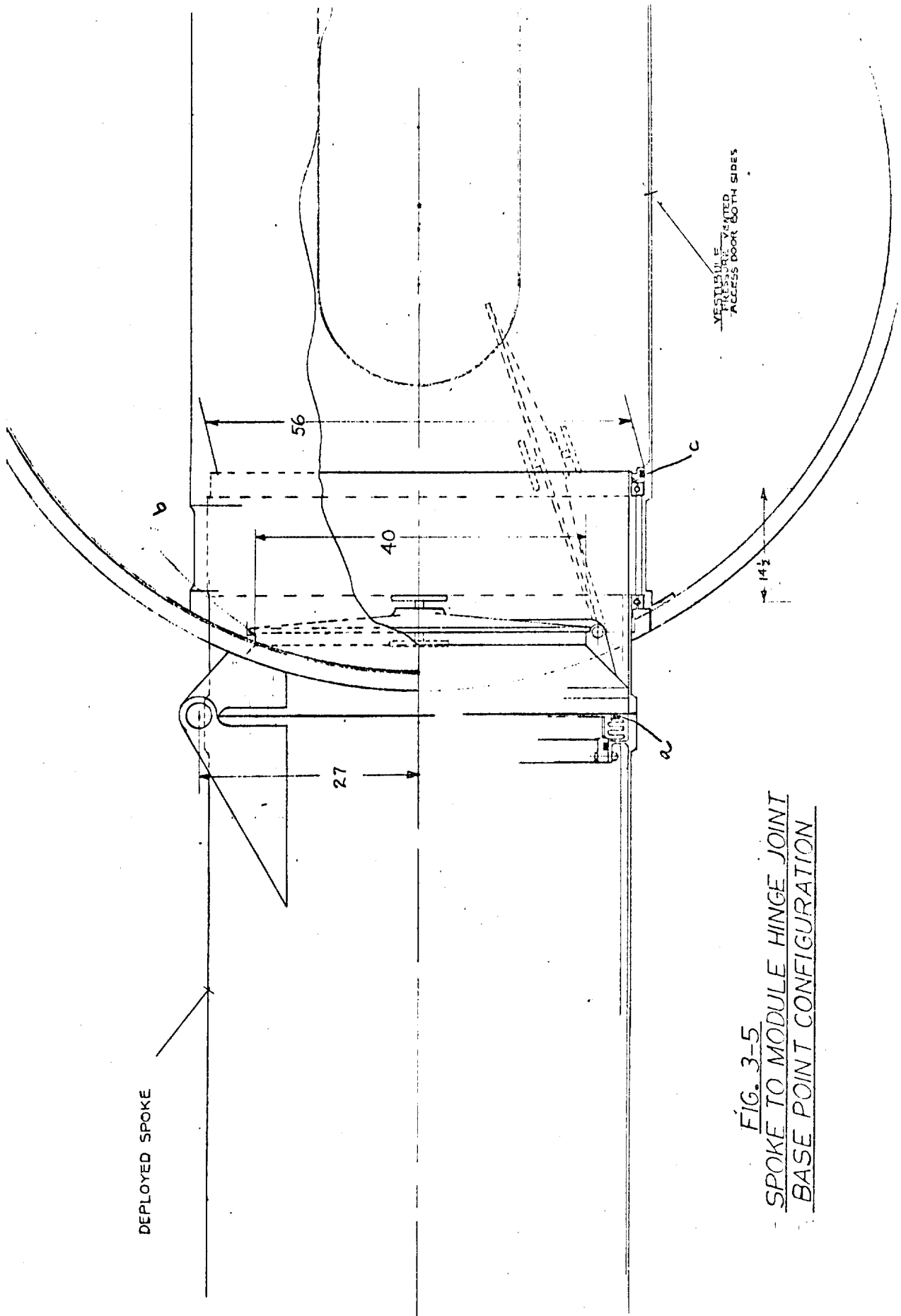


FIG. 3-5.
SPOKE TO MODULE HINGE JOINT
BASE POINT CONFIGURATION

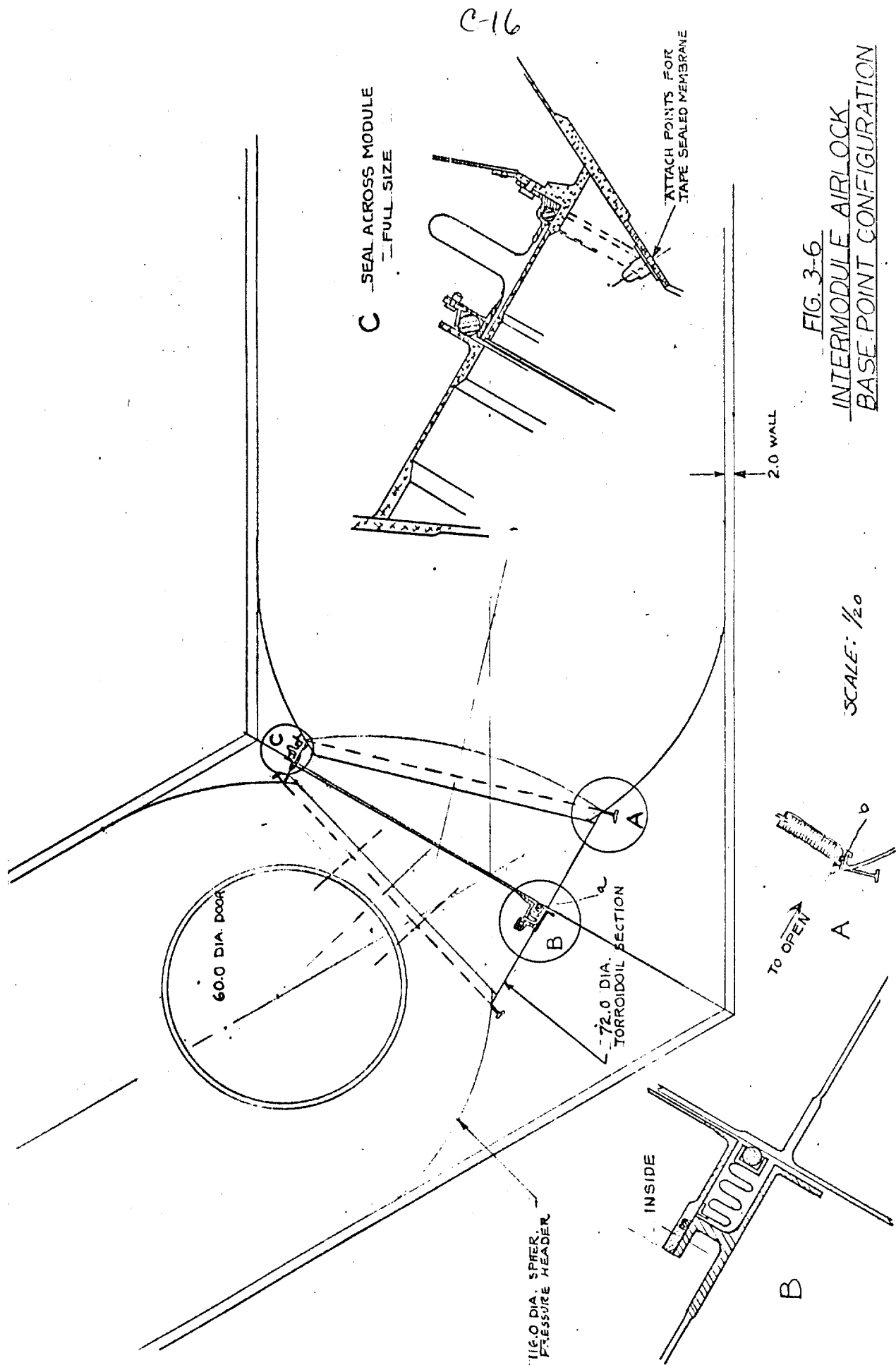


FIG. 3-6
INTERMODULE AIRLOCK
BASE POINT CONFIGURATION

SCALE: 1/20

SEAL MATERIAL ANALYSIS

1. ELASTOMERIC SEALS

An elastomer exposed to high vacuum quickly loses the fluids absorbed from its previous environment as well as volatile compounding ingredients. These compounding ingredients may include low temperature plasticizers, anti-oxidants, antiozonants, antirads and ultraviolet absorbers. Low molecular weight, high polymer materials could also be lost. On the other hand, removal of absorbed oxygen from the elastomer may improve its characteristics. The major effects of vacuum on elastomers is listed in Table D-1.

a. Analysis of 'O' Ring Mechanical Properties

The physical and mechanical properties of an elastomer are functions of the Shore A hardness of the material. Past research has shown that for any given durometer, an average elastic modulus may be determined. Thus the physical reactions may be computed for an elastomer, once the Shore A durometer is obtained. Table D-2 gives the elastic modulus (in compression) and Poisson's ratio of elastomeric material at various Shore A durometers. For the analysis of 'O' rings, the approach taken is through the analogy of elastic bodies, where the seating pressure will be equal to the maximum compressive stress.

The equation for the term S_c is:

$$S_c = .798 \frac{p}{D \left(\frac{1-\delta_1^2}{E_1} + \frac{1-\delta_2^2}{E_2} \right)}$$

Where: p = linear load in lbs per inch

δ_1 = .5 for rubber (Poisson's ratio)

δ_2 = .33 for aluminum (Poisson's ratio)

E_2 = $10.6 (10^6)$ #/in² for aluminum

E_1 = Elastic modulus of the elastomer

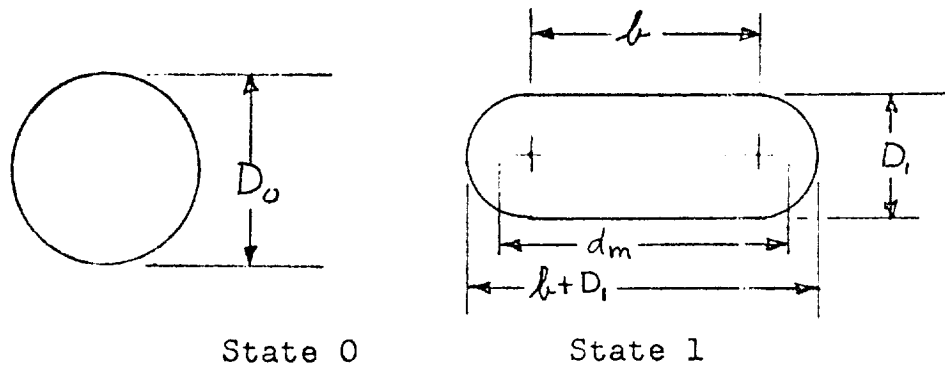
D = 'O' ring diameter in inches

For the purpose of these calculations, it will be noted that the items $\frac{1-\delta_1^2}{E_1}$ and $\frac{1-\delta_2^2}{E_2}$ have a ratio of approximately 1000 : 1, and the second term may be ignored, since the error will be on the order of one tenth of one percent. Thus, to find loading per linear inch for a flange compression joint for an elastomer 'O' ring,

$$p = \frac{S_c^2 D (1-\delta^2)}{.637E}$$

Figure 4-1 shows the values of the function of Poisson's ratio versus various durometers Shore A. To examine the reactions, during compression, of a flange mounted 'O' ring, several assumptions are made. The initial uncompressed state is called 'Zero State', and the final state of compression called 'State 1'.

Area Sections



Assuming circular end areas in the compressed state may introduce a small error, this error will approach zero as the percent deflection increases.

$$A_0 = A_1$$

$$\frac{\pi D_0^2}{4} = \frac{\pi D_1^2}{4} + D_1 b$$

letting $D_1 = aD_0$ then,

$$D_0^2 = a^2 D_0^2 + \frac{4aD_0 b}{\pi}$$

and

$$b = \frac{\pi D_0 (1 - a^2)}{4a}$$

From elastic body theory

$$b = 1.6 \left(p D_0 \frac{1 - \delta_1^2}{E_1} + \frac{1 - \delta_2^2}{E_2} \right)^{\frac{1}{2}}$$

Again, since the terms involving the elastic modulus, and Poisson's ratio of an elastomer and aluminum, respectively, have a ratio of approximately 1000 : 1, the term representing aluminum may be ignored. For convenience, let

$$f = \frac{1 - \delta_1^2}{E_1}$$

Then

$$b = 1.6 [pD_0 f]^{\frac{1}{2}}$$

Equating

$$1.6 [pD_0 f]^{\frac{1}{2}} = \frac{\pi D_0 (1 - a^2)}{4a}$$

$$p = \frac{.240 (1 - a^2)^2 D_0}{a^2 f}$$

Rearranging terms

$$p = \frac{.240 D_0}{f} \left[\frac{1 - a^2}{a} \right]^2$$

Therefore, for a given Shore A durometer, the flange loading may be computed as a function of the diameter, and either the seating stress or the percent deflection. Figures 4-2 through 4-7 correlate these relationships for various Shore A durometers.

1) Radial Loadings

A similar analysis shows that an 'O' ring under a

radial compression will, essentially, be compressed on the diameter upon which the force is applied. In order to make use of the figures mentioned above, care should be taken that the numerical value of the desired compression be doubled before reading. The appropriate Figure will then give the seating stress, and linear load of the seal (for a given diameter and durometer).

b. Analysis of the Mechanical Properties of Flat Gaskets

In many cases flat elastomeric seals and gaskets have been considered as a possible solution for the sealing of joints. The factors analyzed include seating stress, flange loading, and permeation leakage.

Figure 4-8 shows the relationship of percent deflection versus seating stress for various durometers of elastomeric materials. Figures 4-9 through 4-13 present the relationship of percent seal deflection versus flange loading for various durometers and with several width-to-height ratios.

A comparison of pertinent values of a .250 inch diameter 'O' ring of Shore A 60 durometer and the like values of a .250 inch square section flat seal is given in Table D-3.

It can be seen that for the above example, all items

listed are nearly identical, except for flange loading, where the flat elastomeric seal has a value of approximately three times the value of the 'O' ring seal. The Figures show that higher width-to-height ratios will give higher flange loadings for the same thickness, or much lower total deflections for the same mean diffusion path. For the optimum values in any given seal application, it is recommended that 'O' rings be used in all cases.

c. Gas Leakage Through Elastomer Seals

Permeability through a material may be expressed by the following equation:

$$Q = \frac{tA\delta pP}{d}$$

Where: Q = Gas permeation cm^3

t = Time seconds

A = Area normal to permeation flow cm^2

δp = Pressure differential across seal atms.

P = Permeability constant - $\text{cm}^2 \cdot \text{sec}^{-1} \cdot \text{atm}^{-1}$.

d = Width of seal - cm

A tabulation of common elastomeric materials and their respective permeability constants may be found in Appendix III - Table III-1.

For an 'O' ring seal, the path of the permeating gas

is not the diameter. A mean value of that path, d_m may be determined the following:

$$d_m = b + D_1 = b + \frac{\pi D_1}{4} = b + \frac{\pi a D_0}{4}$$

By the mean value theorem of the calculus

$$b = \frac{\pi D_1}{4}$$

But b is also given as follows:

$$b = \frac{\pi D_0}{4a} (1-a^2)$$

$$d_m = \frac{\pi D_0}{4a}$$

From this equation it may be seen that the mean diffusion path is a function of the original diameter and the percentage deflection. Curves for various diameters giving mean diffusion paths versus percent deflection may be found on Figure 4-14.

The form of the original diffusion equation may be altered in the following manner:

$$\frac{dQ}{dt} = \frac{A \delta p P}{d_m}$$

Since the permeability constant is usually given in cgs units, while the dimensions are normally in inches and the permeation leakage is desired in pounds per hour, constants have been derived to make this con-

version. For purposes of these computations

$$1 \text{ cm}^3/\text{sec} = 10^{-2} \text{ \#/hr.}$$

$$\frac{dQ}{dt} = \frac{(3.94 \times 10^{-3}) A \delta p P}{d_m}$$

Where: $\frac{dQ}{dt}$ = Permeation in pounds of air per hour

A = Area normal to permeation (in²)

d_m = Mean diffusion path (in.)

δp = Pressure differential (atm.)

P = Permeability constant (cm²·sec⁻¹ atm⁻¹)

For the purpose of analysis, it may be seen that

$$A = D_1 \text{ times unit length} = a D_0 \text{ times unit length}$$

$$d_m = \frac{\pi D_0}{4a}$$

$$\frac{A}{d_m} = \frac{4a^2}{\pi}$$

which shows that this term, in the permeability equation, is dependent only on the percent deflection and is independent of the diameter of the 'O' ring. Then the rate of permeation per unit length in elastomeric 'O' rings is dependent on only three factors:

- 1) Percent deflection of the 'O' ring
- 2) Pressure differential across the seal
- 3) Permeability constant of the seal material

Since these calculations are based on 10 psia in the interior of the space station, the δp term in this equation is $2/3$. Functional relationships for various permeability constants have been drawn, equating percent deflection against permeation leakage in pounds per hour, and may be seen in Figure 4-15. The mean diffusion path of the flat elastomeric gasket was assumed to be the same as the seating width for the purposes of the analysis. While this is a simplifying assumption, for the deflections under consideration the error will be of a small enough order that it may be neglected. The ability of the seal to absorb and accommodate flexure of the structure, and at the same time have minimum flange loading requirements, dictates the use of 'O' rings in preference to flat section seals.

d. Frictional Forces of 'O' Rings

Experimental determinations of the coefficient of friction using an elastomer surface against aluminum with a micro-finish of (3) is about (.47). This is a coefficient of sliding friction. Numerous recorded experiments show that the static or 'break out' friction will vary from two to three times the static friction, in cases of rubber opposed to smooth metal.

Thus the value of static friction for 'O' rings used in a piston or sleeve will be between (.94) and (1.41).

It is obvious that these coefficients must be carefully considered when computing actuation forces. The running friction, while it contributes to the forces during the sliding operation of the passageways, is lower than the static values, and therefore the sum of the static forces will yield maximum values. This higher force, while it is only momentary in nature, will dictate the requirements for actuation. Since the frictional loads are independent of the surface areas, but depend only upon the forces normal to the surfaces, these loads may be determined for any seal by finding the load per linear inch of the specified 'O' ring from the Figures, and multiplying this by the frictional coefficient. It should be noted that in using the Figures, a direct reading of percent deflection cannot be used. These graphs were for compression between two plates. If the percent deflection on one side of an 'O' ring is to be 10%, then read 20% on the graph to obtain flange loading for a sliding 'O' ring. For purposes of computing maximum

frictional loads, a coefficient of 1.4 has been used.

2. ANALYSIS OF METALLIC SEALING RINGS

Applicable commercially available design of large diameter metallic gaskets are based on a 'C' or 'E' configuration. This allows for a maximum amount of deflection, while still retaining a local sealing pressure. Since there is, effectively, no sealing assistance from air pressure to expand the 'C' or 'E' ring, a rigid type of flange must be employed to obtain a relatively constant sealing force around the periphery of the seal, when taking into consideration the deflection ratio of the selected sealing ring, Figure 4-16.

The basic material for standard sealing rings is usually 302 stainless steel, with aluminum used when lower flange loadings are desired. Since a surface finish defect would invalidate the seal for all practical purposes, it is highly desirable to coat the sealing surface with some material which would make allowances for the slight surface imperfections in the opposing surface.

Five materials are suggested, with some of their limitations, as suitable for this application.

- 1) Silver Plate, about .001 to .0015 inches
- 2) Teflon 100, about .001 to .002 inches
- 3) Polyethylene (high density), about .001 to .002 inches

- 4) Hard Rubber, about .002 to .003 inches, (such as Parker Seal 77-545, V271-7 or Viton A)
- 5) Lead, about .001 to .002 inches

It should be noted that Silver Plate tends to embrittle in a vacuum environment and Teflon 100 degrades under continued radiation.

These materials all have low evaporation rates in a hard vacuum, as well as low permeability constants. These permeability constants are not a large factor when the area normal to the mean path of gas flow is considered. Within required temperature ranges, the physical characteristics are, in general, adequate. The deformation of the coating materials themselves will offer a surface of contact, which will protect the opposing surface from scratching, and assure a hard vacuum seal.

The selection of an interface seal should be made after due consideration of the following:

- 1) Retention of seal resiliency
- 2) Seal interface leakage
- 3) Seal permeability
- 4) Resulting structural loading
- 5) Environmental effects on seal material

In considering the use of metallic 'C' or 'E' rings, with or without a surface coating, several of the points listed above give superior performance. Through judicious choice of base material, such as 302 stainless steel, or aluminum coated with lead, these requirements can be met.

The resulting structural loadings, using a metallic base material, are extremely high. They will run from 10 to 20 times the loads for equivalent elastomeric 'O' ring seals. This in itself, should be a sufficient reason to eliminate the metallic ring as a desirable seal. Figures 4-16, 4-17 and 4-18 depict the required seating stresses for the various metallic gasket configurations. The additional structural requirements, and accompanying increase in weight, would overshadow the effect of permeation leakage of air through elastomeric seals.

Table D-1

THE MAJOR EFFECTS OF VACUUM ON ELASTOMERS

1. Exposure to high vacuum does not effect the mechanical properties unless high temperatures are also present.
2. The flexibility of elastomers exposed to low temperatures is negatively effected by high vacuum exposure.
3. Large weight losses generally only accompany elastomers with high levels of plasticizer.

Table D-2

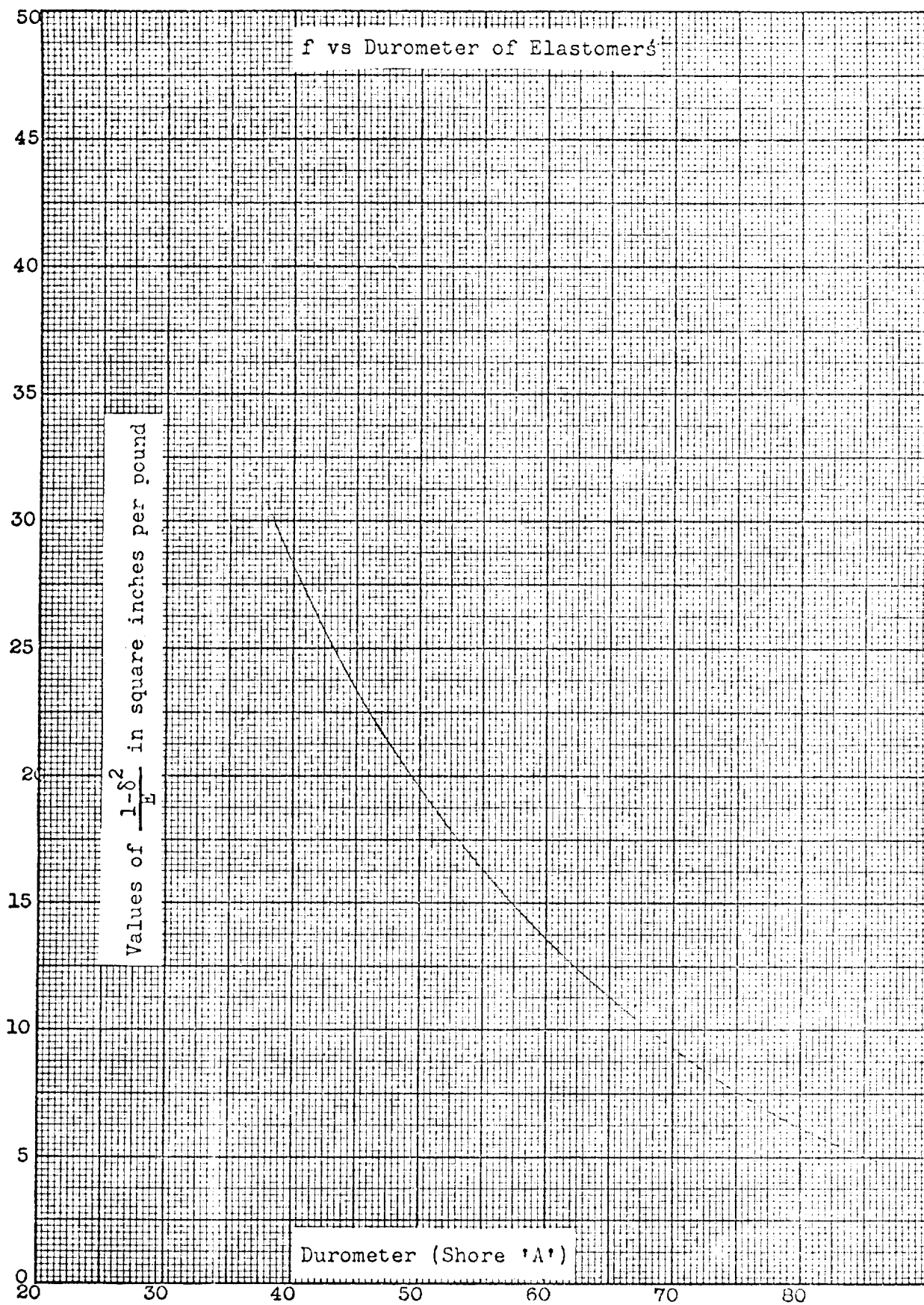
f FACTOR vs. DUROMETER FOR ELASTOMERS

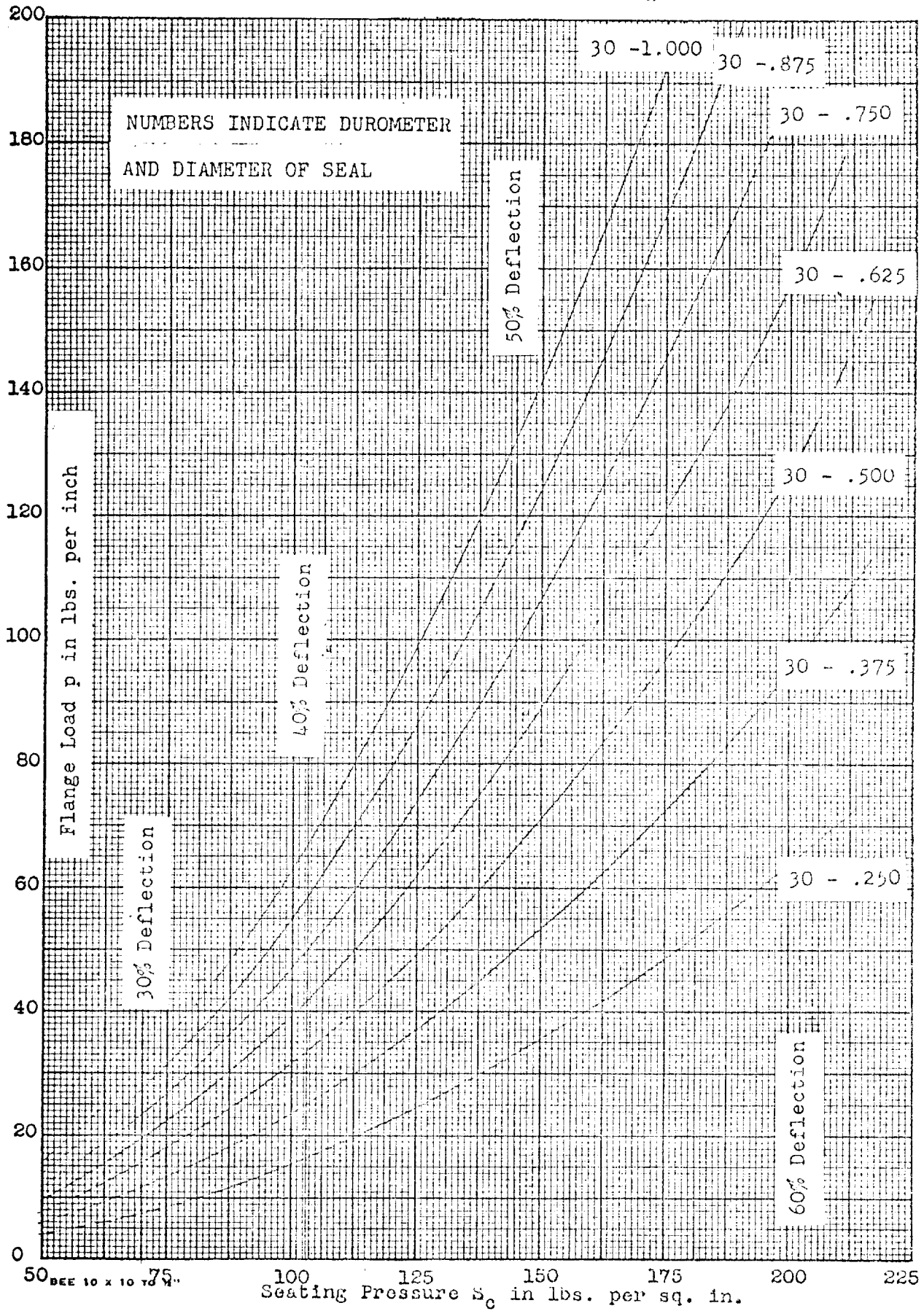
SHORE 'A'	ELASTIC MODULUS (#/in ²)	$\frac{1-\delta^2}{E}$ (in ² /#)10 ⁻⁴
30	185	40.5
40	275	27.3
50	380	19.7
60	535	14.0
70	740	10.1
80	1200	6.25

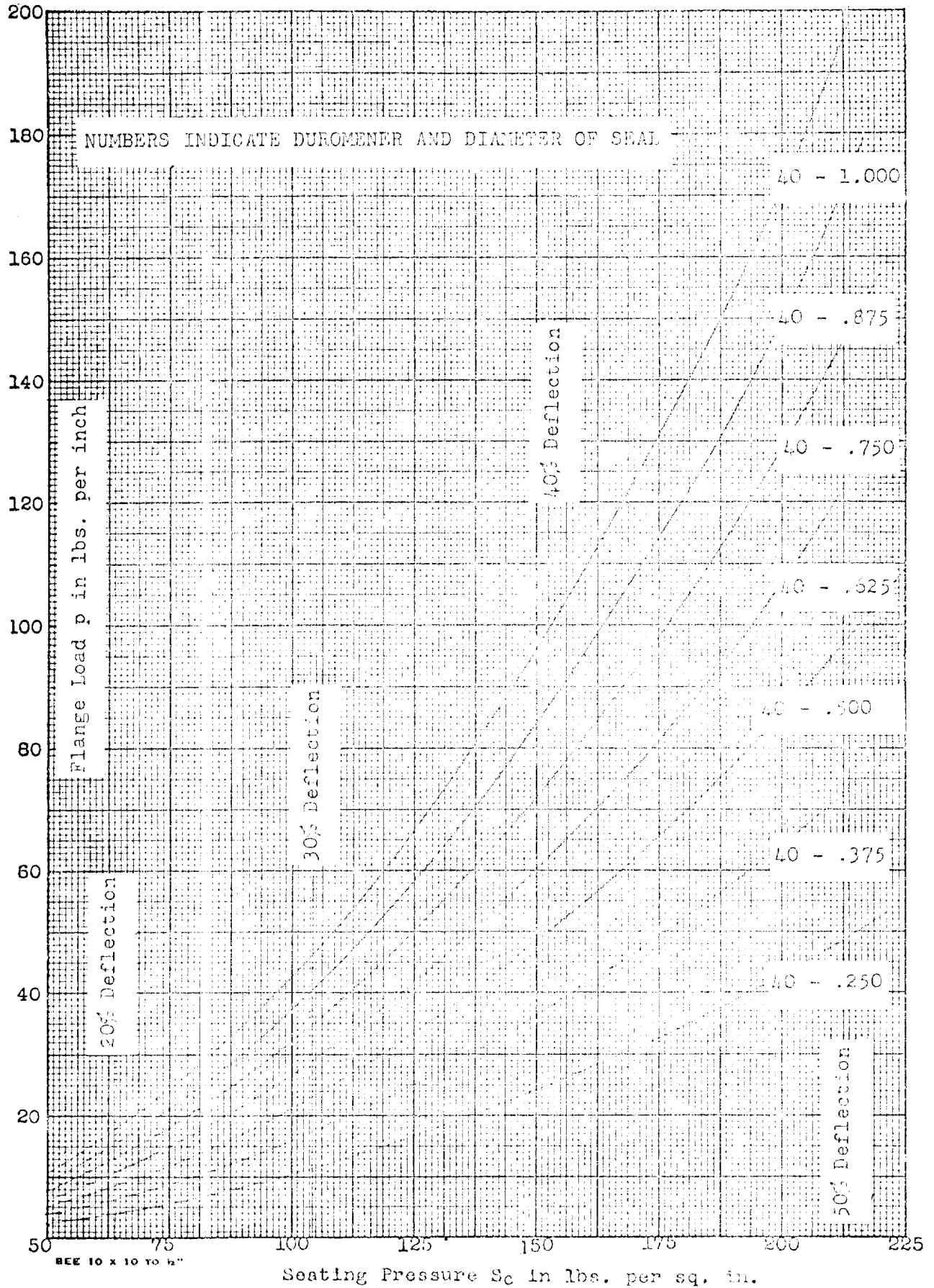
Table D-3

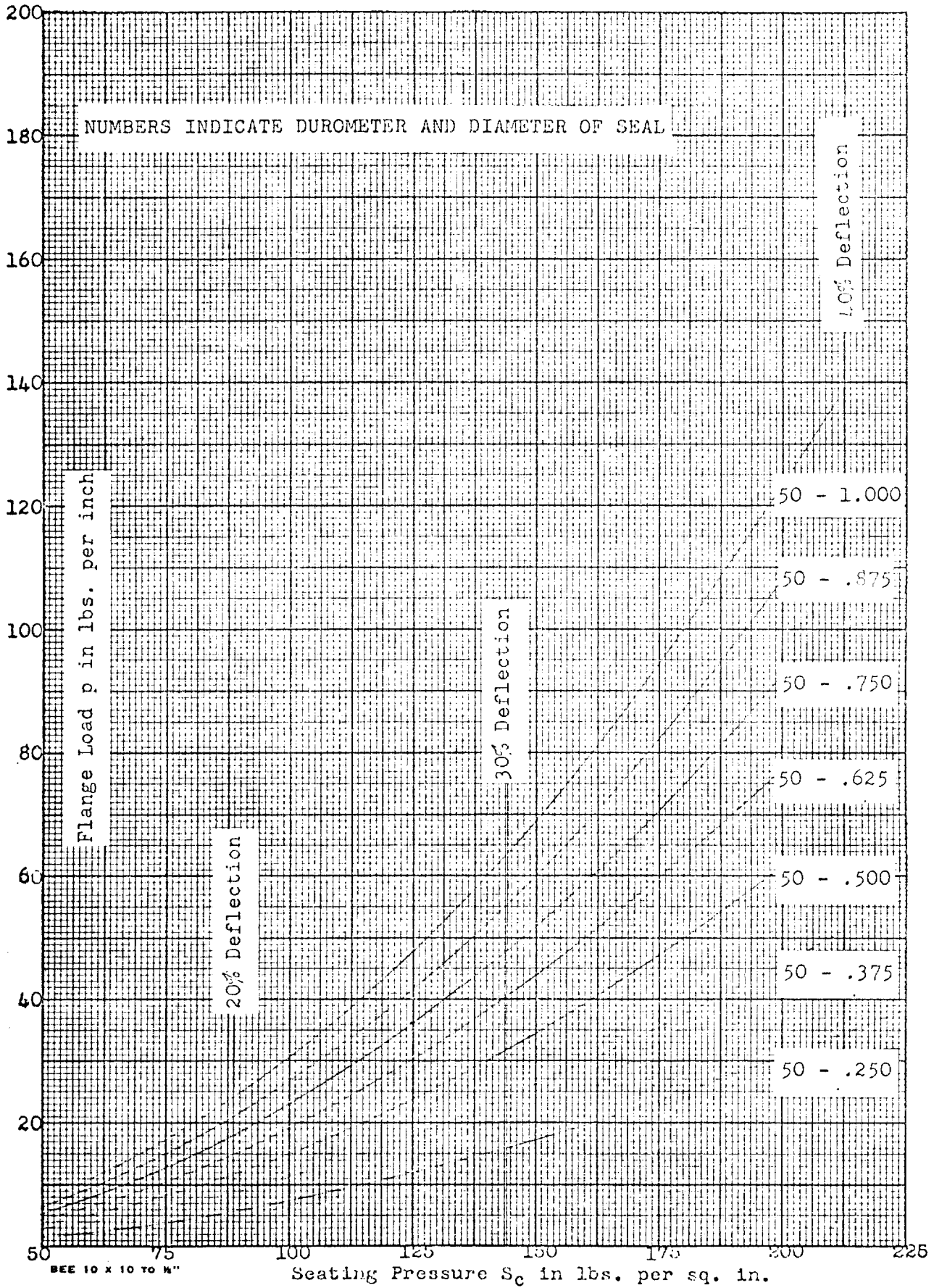
COMPARISON OF 'O' RING AND FLAT GASKET

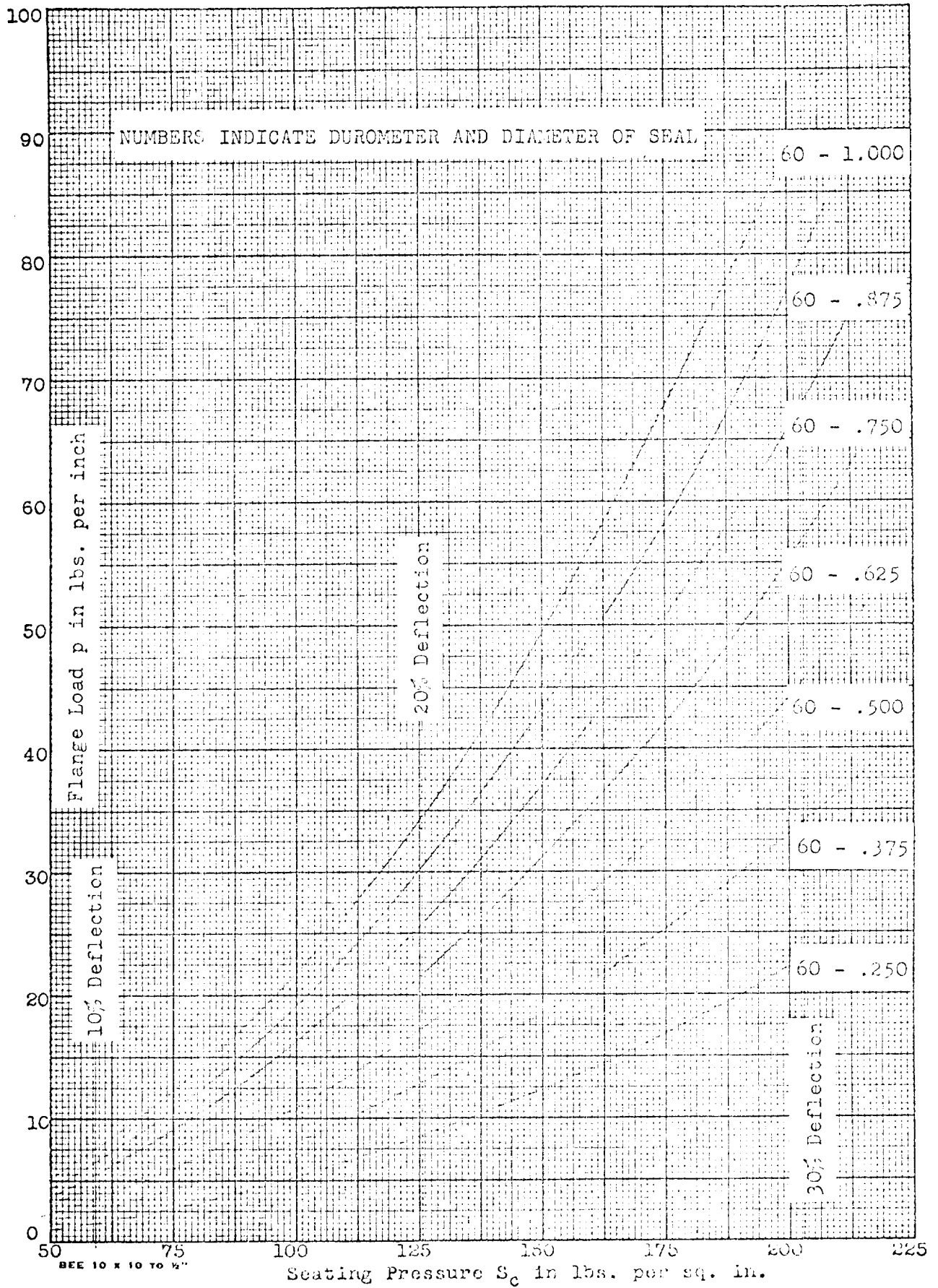
<u>PARAMETER</u>	<u>'O' RING</u>	<u>FLAT GASKET</u>
Durometer	60 Shore A	60 Shore A
Deflection	20%	20%
	.05 inches	.05 inches
Mean Diffusion Path	.25 inches	.25 inches
Seating Stress	125 psi	108 psi
Flange Load	8.5 #/in.	27.0 #/in.

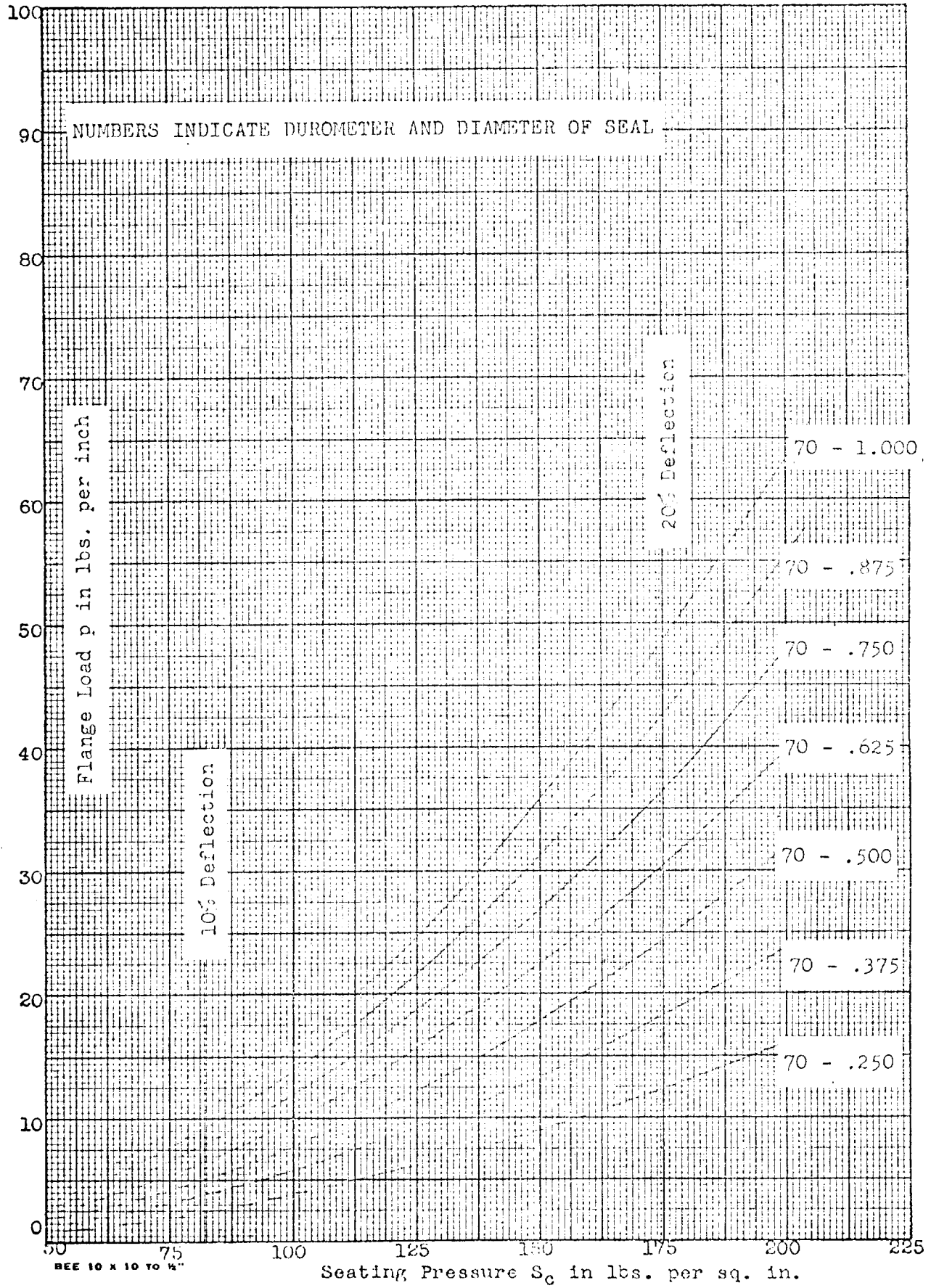








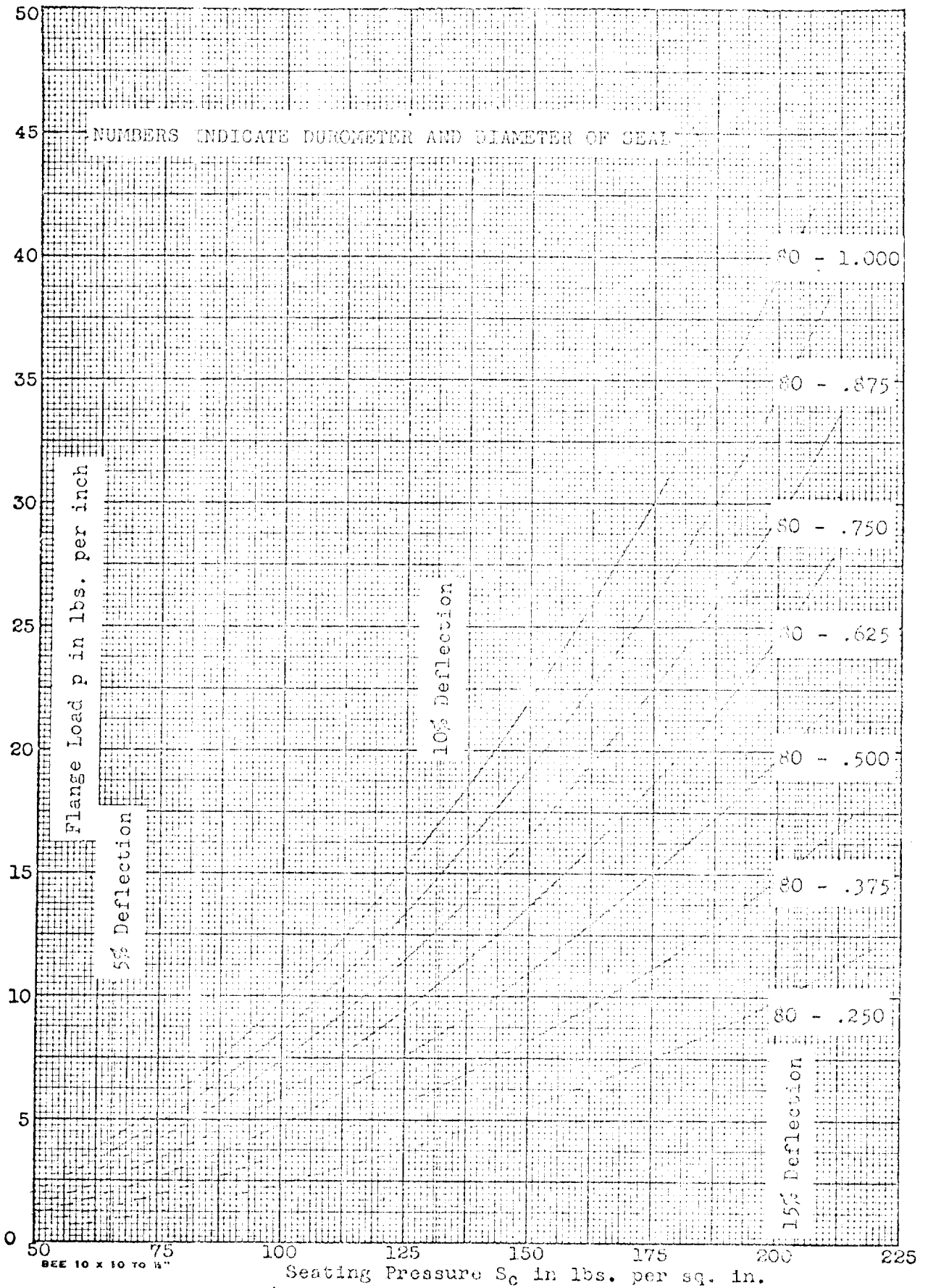




D-22

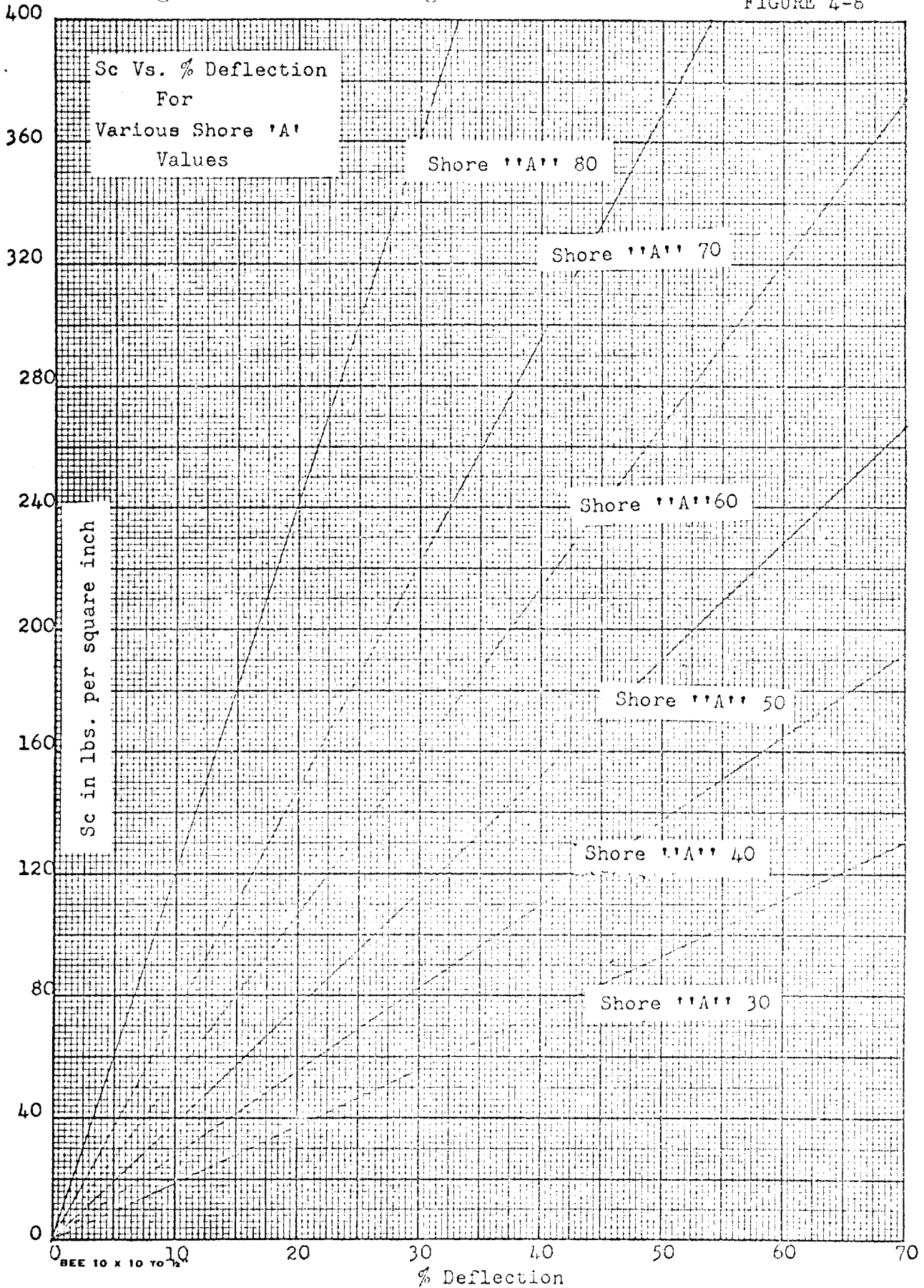
S_c vs. p for 80 Durometer "O" Ring Seals

FIGURE 4-7

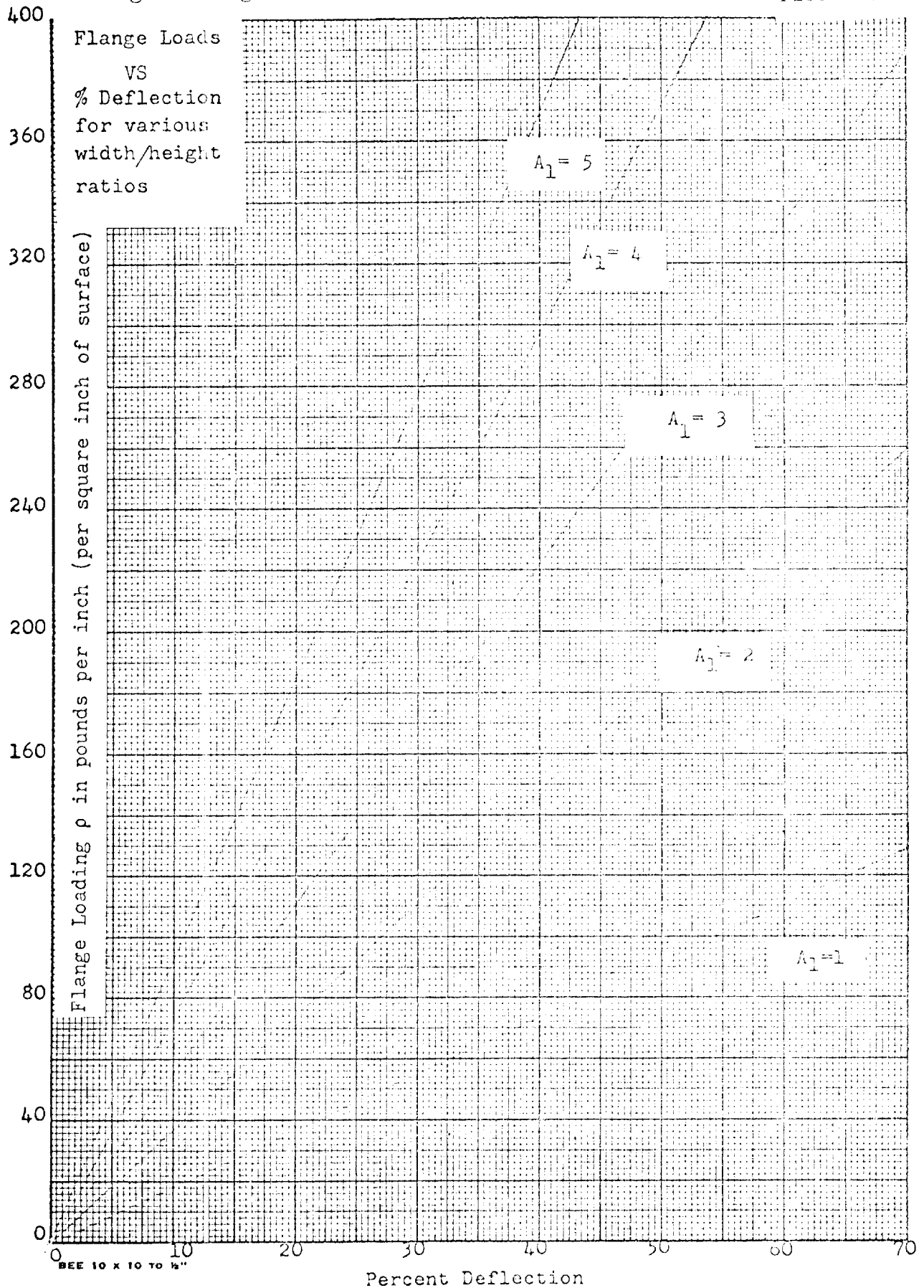


Seating Stress for Flat Flange Elastomer Seal

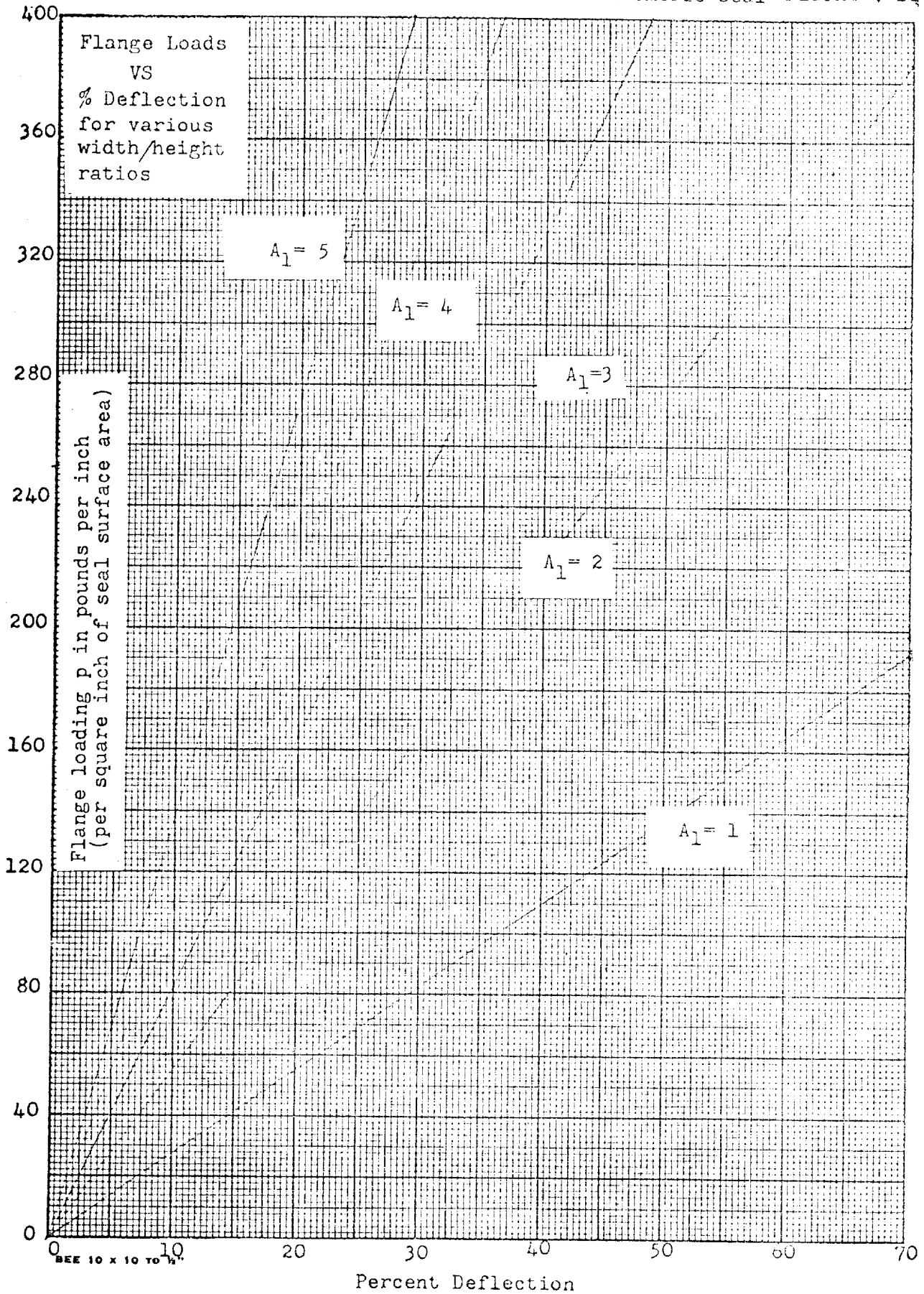
FIGURE 4-8



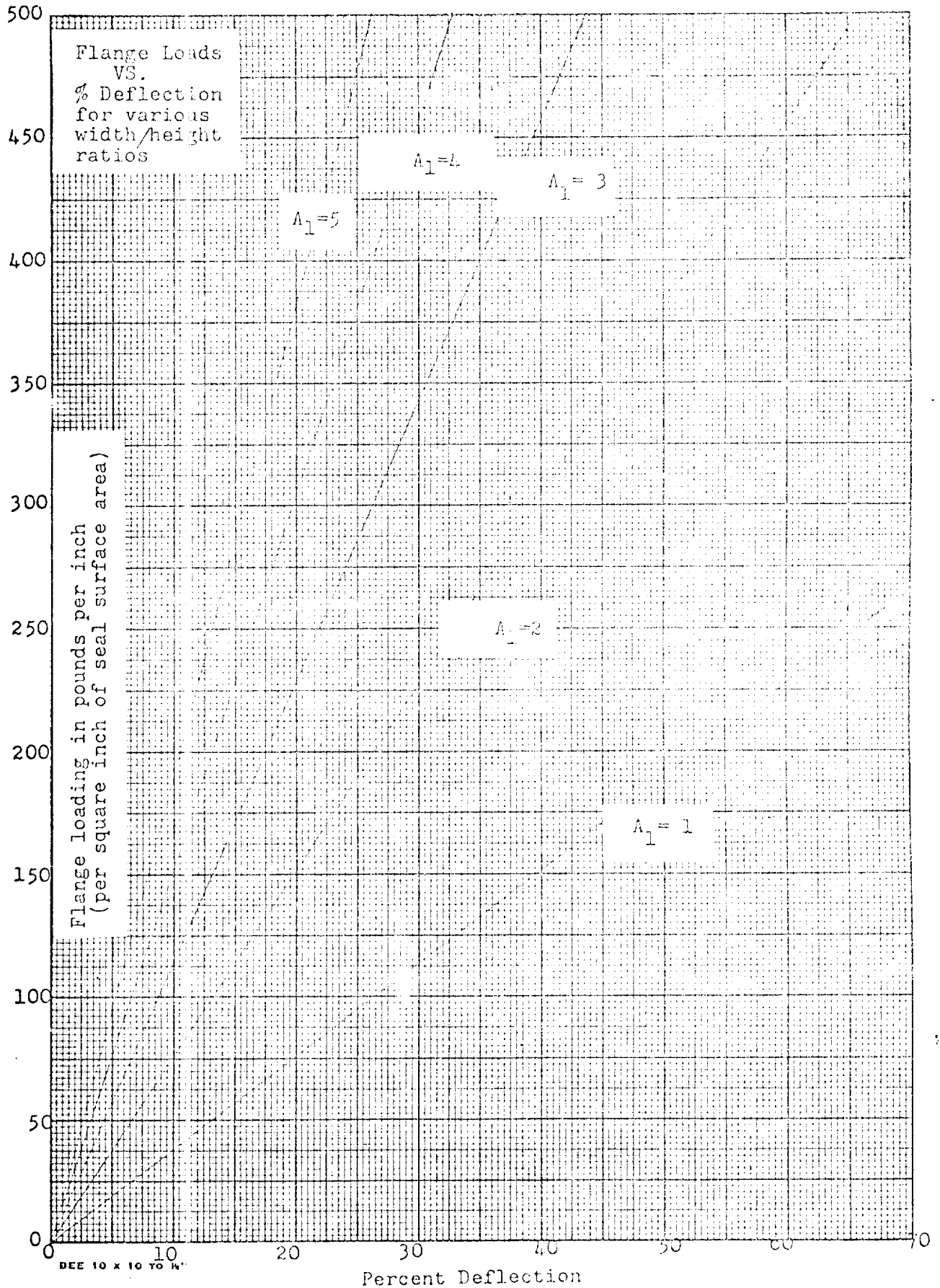
Flange Loading for Shore "A" 30 Flat Elastomeric Seal FIGURE 4-9



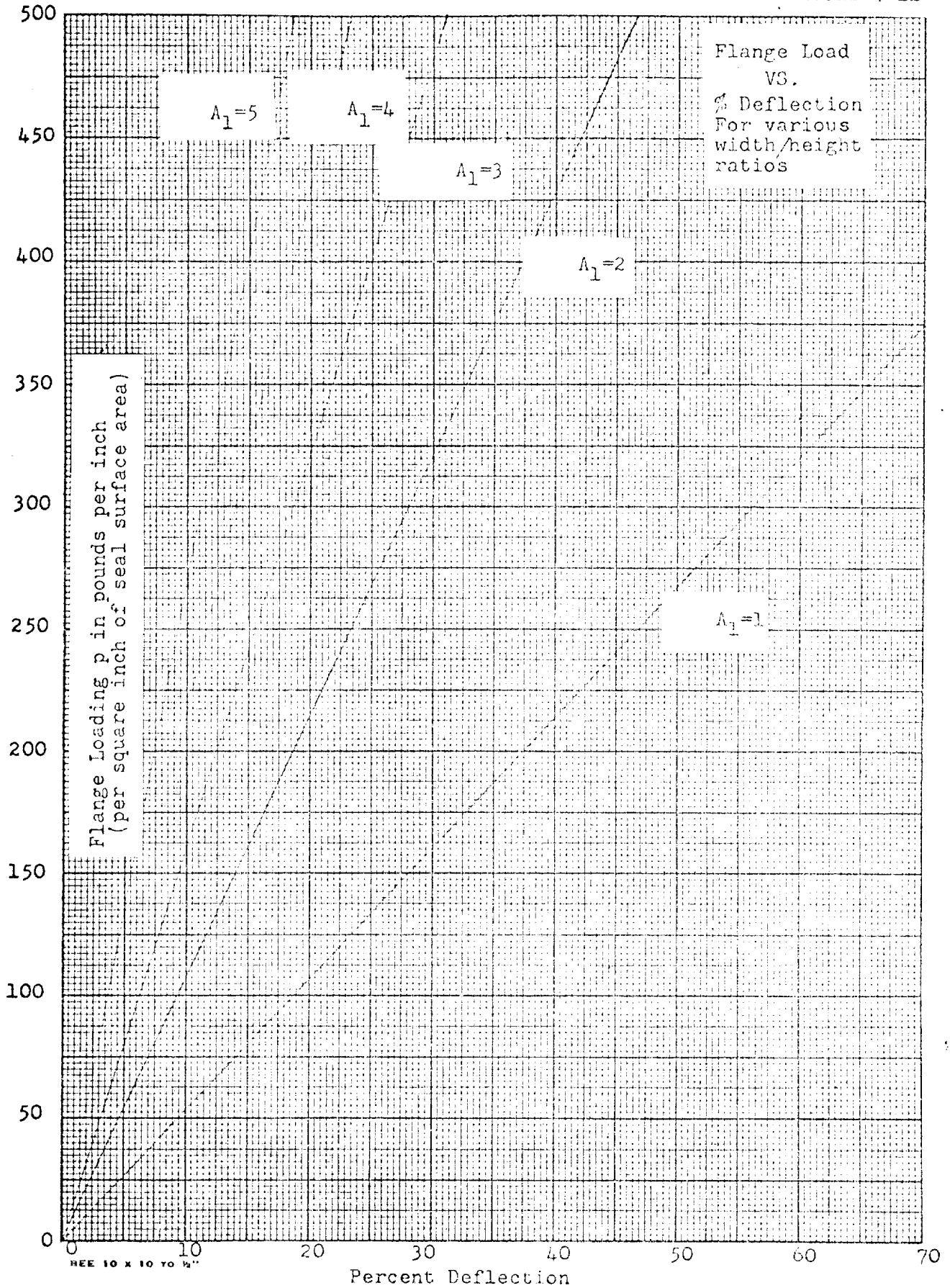
Flange Loading for Shore "A" 40 Flat Elastomeric Seal FIGURE 4-1Q



Flange Loading for Shore "A" 50 Flat Elastomeric Seals FIGURE 4-11

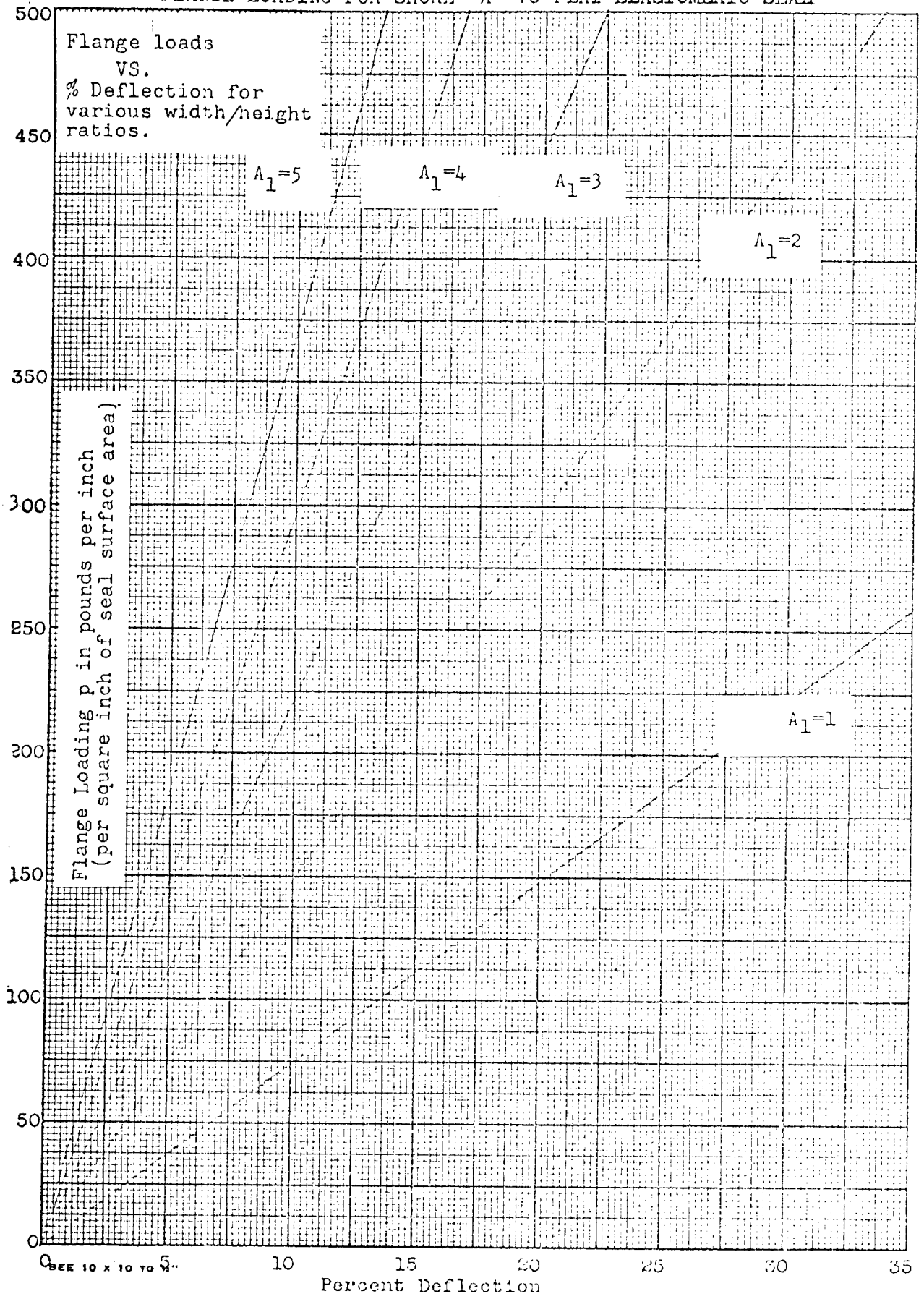


Flange Loading for Shore "A" 60 Flat Elastomeric Seal FIGURE 4-12



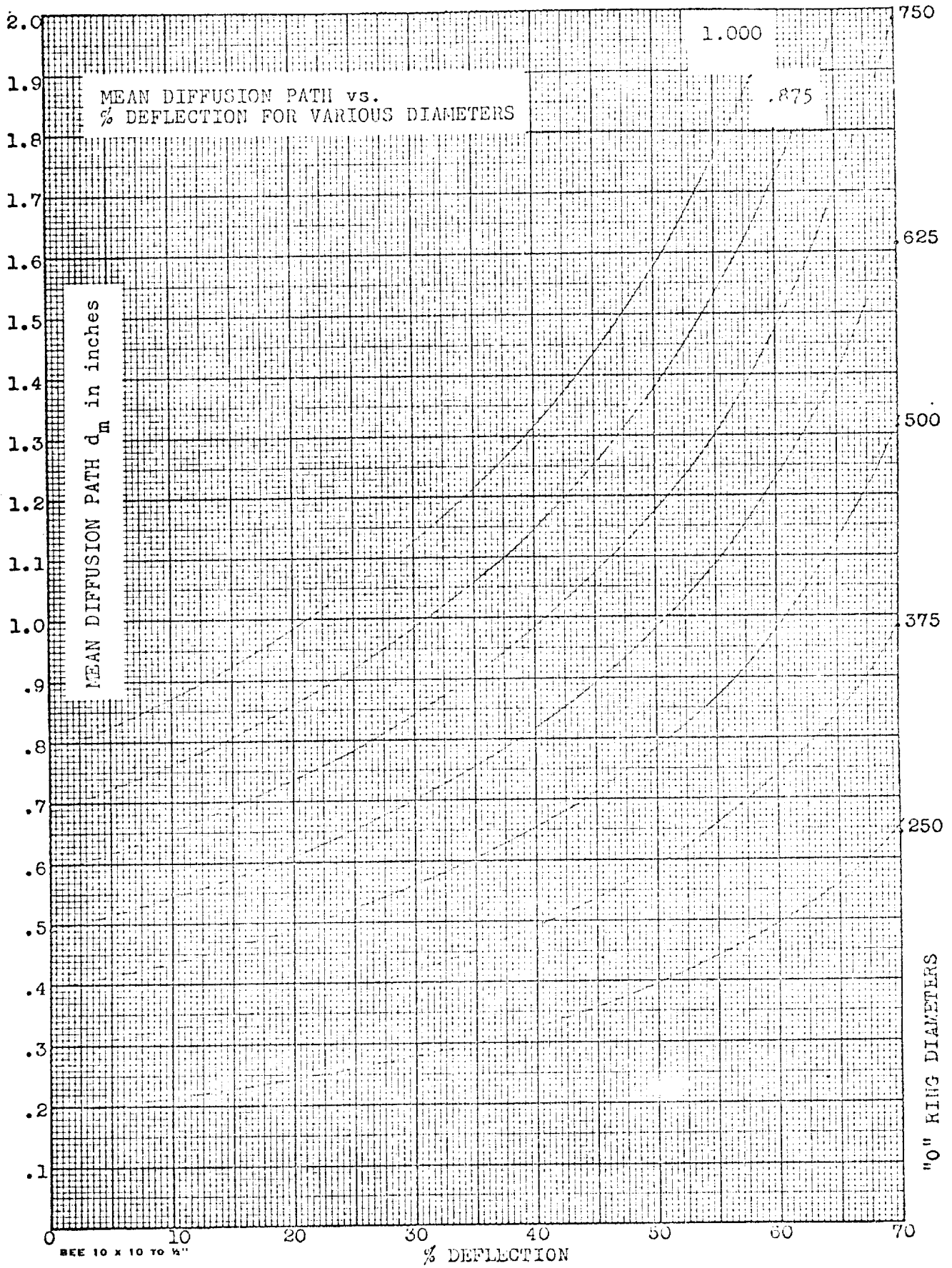
FLANGE LOADING FOR SHORE "A" 70 FLAT ELASTOMERIC SEAL

FIGURE 4-13

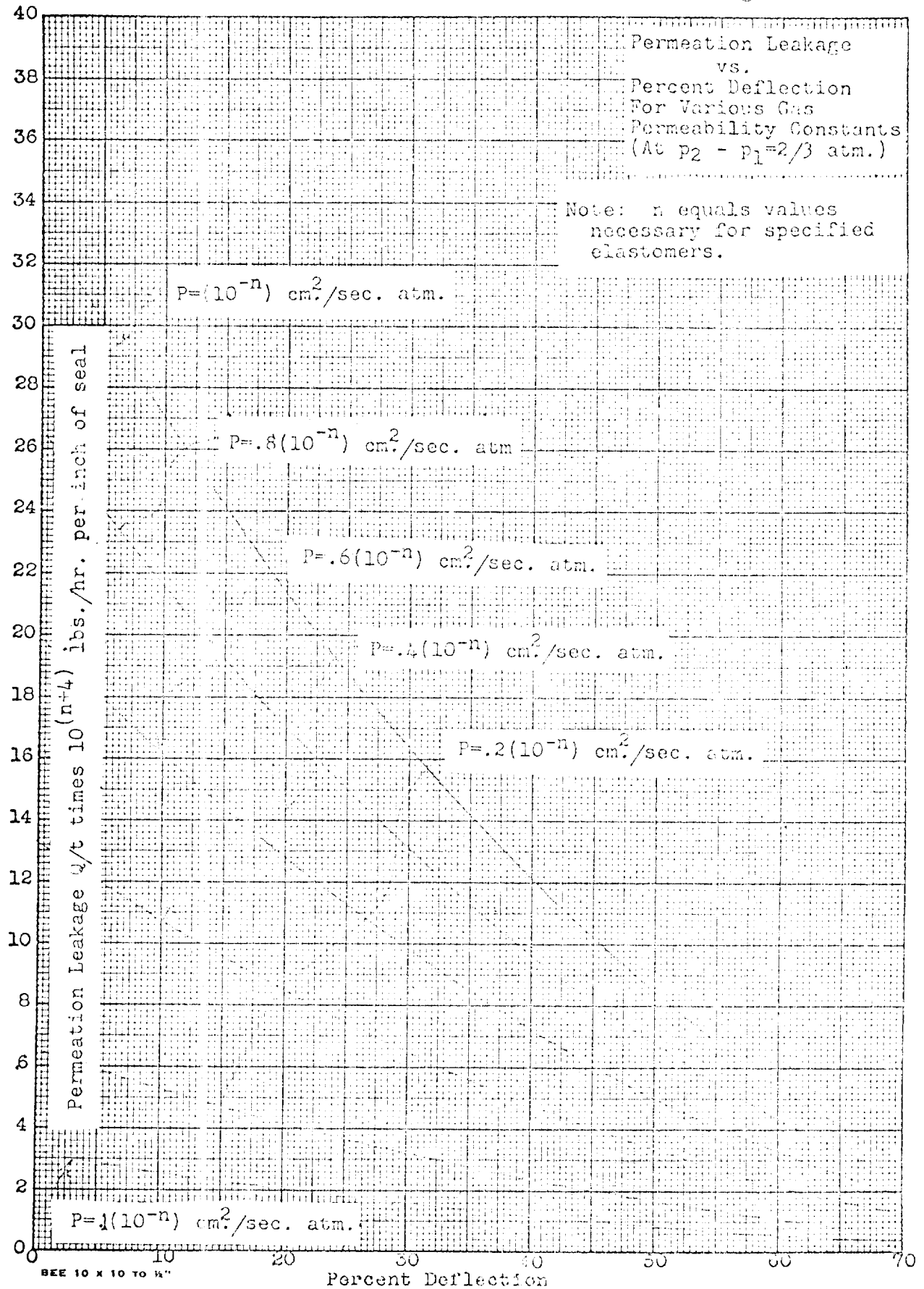


MEAN DIFFUSION PATH OF GAS THROUGH "O" RINGS

FIGURE 4-14



Permeation Leakage for Elastomer "O" Rings FIGURE 4-15






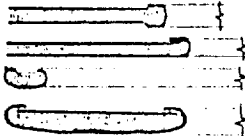

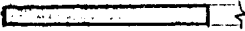

TYPE	MATERIAL	THICKNESS (in.)	MIN. SEATING STRESS
	Aluminum		1500 psi
	Copper		2000
	Soft Steel (Iron)	1/8	4000
	Monel		4500
	Stainless Steel		6000
	Aluminum		2000 psi
	Copper		2500
	Soft Steel (Iron)	1/8	3000
	Monel		3500
	Stainless Steel		4000
	Carbon Steel	1/8	2500-15,000 psi
	Carbon Steel	3/16	2500-15,000
	Stainless Steel	1/8	3000-30,000
	Stainless Steel	3/16	3000-30,000
	Lead		500 psi
	Aluminum		2500
	Copper		4000
	Soft Steel (Iron)	1/8	6000
	Monel		7500
	Stainless Steel		10,000
	Lead		500 psi
	Aluminum		1000
	Copper	approx.	2500
	Soft Steel (Iron)	9/64	3500
	Monel		4500
	Stainless Steel		6000
	Aluminum		16,000 psi
	Copper		36,000
	Soft Steel (Iron)	1/8	55,000
	Monel		65,000
	Stainless Steel		75,000
	Aluminum		20,000 psi
	Copper	1/32	45,000
	Soft Steel (Iron)	and	68,750
	Monel	1/16	81,250
	Stainless Steel		93,750

FIG. 4-16

MINIMUM GASKET SEATING STRESS





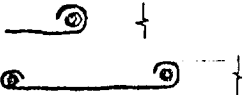
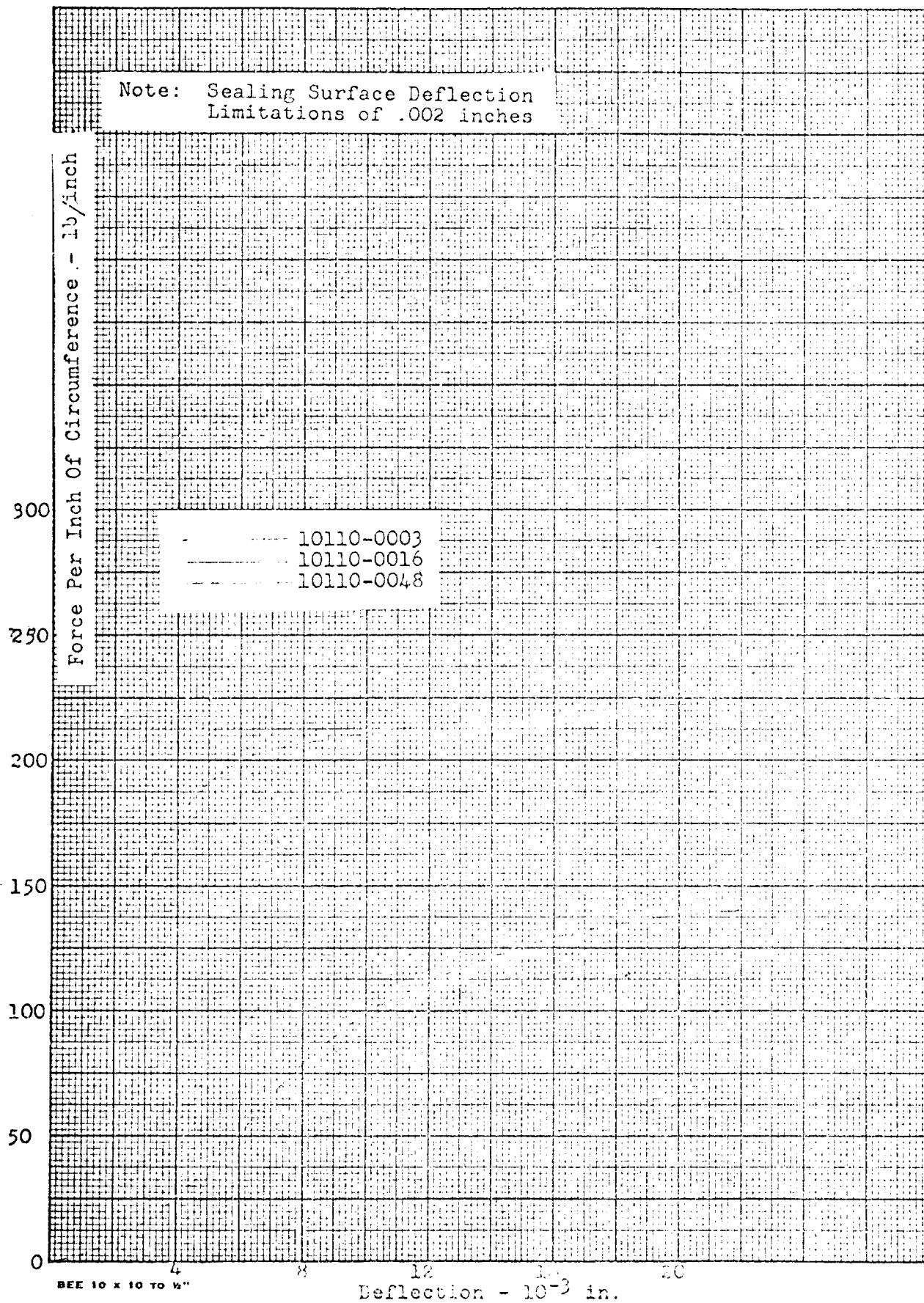
	MATERIAL	THICKNESS	MIN. SEATING STRESS
	Aluminum		25,000 psi
	Copper	1/8 pitch	35,000
	Soft Steel (Iron)	all thicknesses	55,000
	Monel		65,000
	Stainless Steel		75,000
	Aluminum		30,000 psi
	Copper	1/16 pitch	40,000
	Soft Steel (Iron)	all thicknesses	60,000
	Monel		70,000
	Stainless Steel		80,000
	Aluminum		35,000 psi
	Copper	1/32 pitch	45,000
	Soft Steel (Iron)	all thicknesses	65,000
	Monel		80,000
	Stainless Steel		95,000
	Aluminum		1300 lb/circular in
	Copper	Any diam	4500 lb/circular in
	Soft Steel (Iron)		6000 lb/circular in
	Stainless Steel		
	Aluminum Jacket	Any diam	1500 lb/circular in
	Aluminum Cores		
	Aluminum Jacket	Any diam	1500 lb/circular in
	Stainless Steel Cores		
	Stainless Steel Jacket	Any diam	6000 lb/circular in
	Stainless Steel Cores		

FIG. 4-16 (Cont'd)

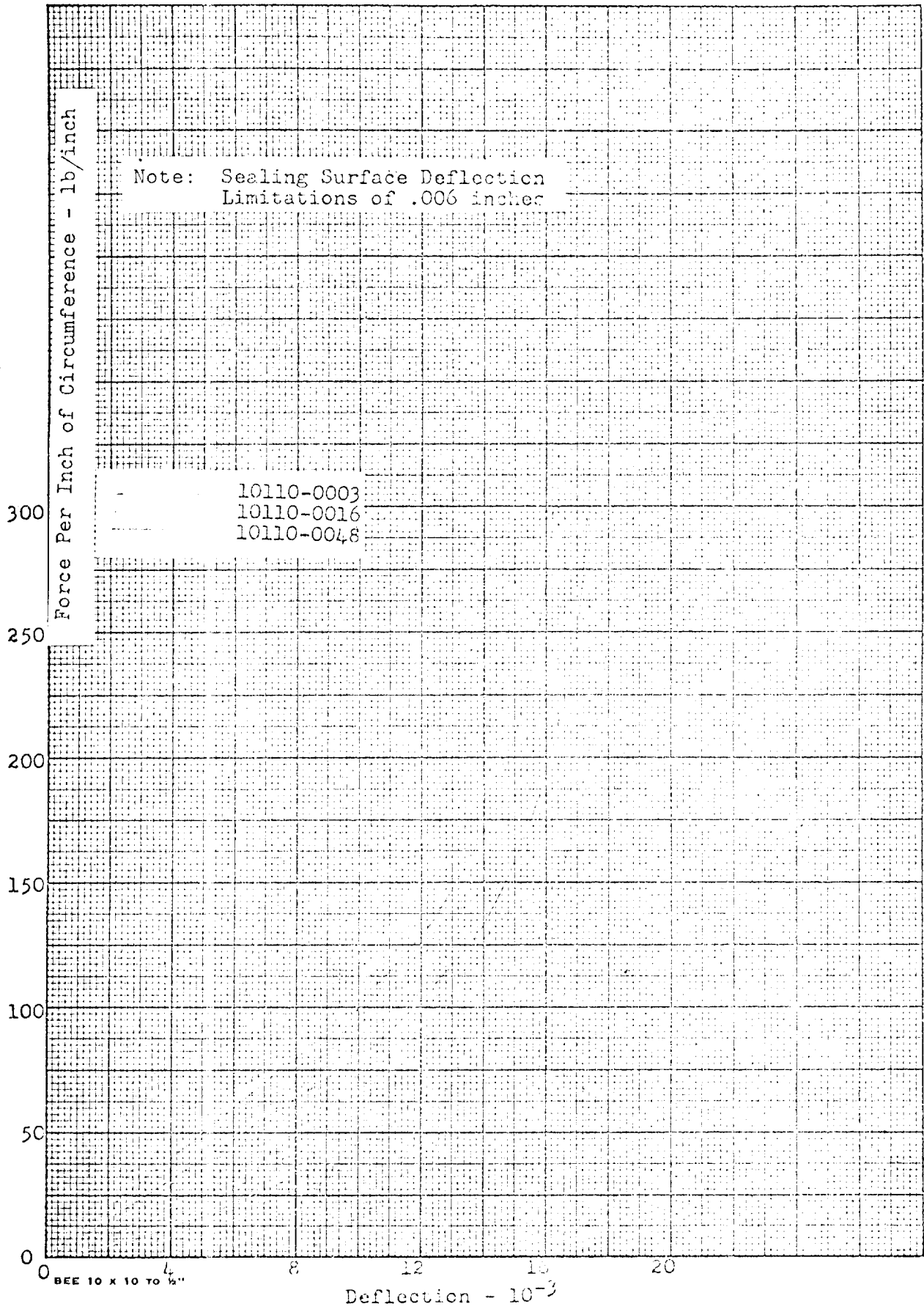
FORCE vs. DEFLECTION CURVES C-RING SEAL

Fig. 4-17



FORCE vs. DEFLECTION CURVES E-RING SEAL

Fig. 4-18



RESULTANT DESIGN CONCEPTS

In the previous sections the basic connection-seal design parameters and operating criteria have been developed. This has yielded a definitive picture of the contribution of seal seating stress to connection clamp and flange design, of the modifications in design necessary to accommodate station start-up by the astronauts, and the representative section parameters of the base point configuration. This section depicts various modifications to the base point configurations required to optimize weight and reliability and further analyzes these modified concepts to yield section parameter envelopes.

1. DOCKING ACCESS AIRLOCK

a. General Description

As stated previously, an initial airlock must be provided to allow personnel entrance into the station from the Apollo vehicle. To enable the astronauts to enter the station without traversing through free space, the airlock functionally comprises an extendable passageway. In the base point design (reference Section C), this movable passageway was designed into the hub.

This placement engenders the following operational problems.

The 16 inch travel of the passageway comprises on-board station space.

Replacement of the sliding seals is difficult if not impossible.

The seals are called upon to be operational for the full two-year station life.

Separate hydraulic actuators must be provided to facilitate passageway travel.

A logical design modification was synthesized by utilizing the extending airlock presently planned for the Apollo resupply vehicle while considering the operational procedures contemplated for the station.

Current plans for the Apollo vehicle include an airlock section which extends some 15 inches into the Apollo vehicle at full inward travel. This technique allows the crew to enter the passageway while the nose cap of the Apollo is closed and further allows the airlock to be sealed off to prevent loss of vehicle air.

This planned Apollo concept was modified to yield two-way travel of the passageway, so that during

docking, the separation between the Apollo and the station can be bridged by the passageway. The following major advantages are derived from this placement.

All seals except the hub access door have been shifted to the Apollo portion of the lock. These seals can be checked in the Apollo vehicle before use.

Failure of the sliding seals does not cause total station depressurization, since the leak can be sealed off by closing the hub access door. The operational life requirement of the major portion of the seals can be reduced to 6 weeks in the case of the base point design.

Airlock linear travel can be accomplished with air pressure, using bottled air carried up by the Apollo, thus eliminating the need for auxiliary hydraulic actuators.

b. Extending Airlock

Figures 5-1, 5-2, and 5-3 depict conceptual configurations of an extending airlock.

The operational cycle takes the following form.

The passageway interlocks are released (the Apollo nose cap being in the closed condition).

Controlled gas flow is fed to the passageway to provide a pressure slightly in excess of vehicle pressure. (this results in a controlled travel of the Apollo portion of the passageway inward to the Apollo vehicle)

The extension interlocks are activated, locking the Apollo passageway section upper structure. The astronaut enters the airlock via the access door and checks the seals by visual inspection. The Apollo nose cap is opened.

Under astronaut control the outer passageway section interlocks are released and air pressure is bled in a controlled manner from the annular pistons, allowing the outer and inner coupled passageway sections to traverse the intervening 19 inch free-space region.

The internal vehicle pressure is used as the force mode.

The Apollo-hub mating seal is made up during the final four inches of travel and comprises a circumferential seal which linearly enters the tapered entry section and seal area on the station hub.

c. Folding Access Door

Figure 5-4 depicts a folding access door for use in conjunction with the extending passageway. This type door provides a temporary seal which is active during the unlocked portion of the docking maneuver. The door is opened into the passageway after airlock make-up and provides a maximum free area.

Another method considered was to eliminate the door altogether, and with this modification, initial station entry takes the following form.

All functions will be the same as above except that the astronaut will not be required to enter the passageway prior to connection.

After the initial docking latches are activated, controlled travel of the depressurized passageway is initiated. Upon contact, a visual inspection through the hatch visual access would be made and initial pressurization of the passageway would occur.

It should be noted that this design significantly reduces the weight required for the sliding passageway.

If no leakage is evidenced, the astronaut then proceeds through the two hatches.

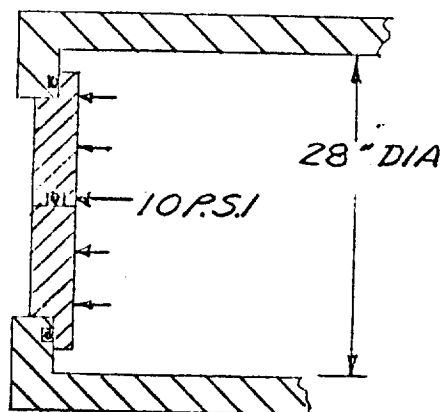
Withdrawal of the passageway during subsequent docking could be accomplished with auxiliary actuators.

d. Hub Access Door

Figures 5-5, 5-6 depict the suggested configurations for the hub access door. The hub access door utilizes a ball-lock type latch capable of being opened from either side. The ball lock is provided to attain maximum load distribution while keeping the frictional contact area at a minimum. An automatic interlock can be naturally provided by controlling the distance of travel of the passageway. Figure 5-5 shows that full extension of the passageway automatically unlocks the access door.

e. Representative Stress and Sizing Calculations

1) Passageway Access Door (Folding Type)



The loading on the door consists of 10 psi internal pressurization.

$$A = \text{Area of Door} = \pi r^2 = 615 \text{ in}^2$$

$$\text{Circumference} = \pi D = 88 \text{ inches}$$

$$\text{Load} = P A = 6,150 \text{ lbs.}$$

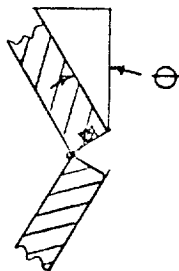
The unit flange loading is then:

$$\frac{6150}{88} = 70 \text{ lb/in load on flange}$$

Consider a circular cross section elastomer seal at the center hinge; use 40 durometer at 20% deflection, then,

$$\begin{aligned} \text{Flange Load} &= 4 \text{ lbs/in (.250 dia. 'O' ring)} \\ &= 112 \text{ lbs (radial load - closed)} \end{aligned}$$

The closing force perpendicular to the door is a function of the sine of the angle θ and the deflection of the 'O' ring at that angle.



This force will remain very small since sine θ decreases as deflection of 'O' ring increases. The door will be secured by four equally spaced clamps. Considering the door as a clamped edge flat plate, the moment at the edge equals M_r (in-lbs/in).

$$M_r = \frac{-pa^2}{8} = 245 \text{ in-lbs/in}$$

Consider a .375 dia. 'O' ring around door, 50 durometer, with 20-30% deflection, then the flange load is 18 lbs/in.

$$\text{Clamp load} = 396 \text{ lbs/clamp}$$

These loads are so small magnitude and a relatively light clamp design will easily handle them.

$$\Sigma_{\max} = \frac{3pa^2}{4h^2} = \frac{1470}{h^2}$$

Therefore the thickness of the door may be kept extremely small, based mainly on the physical inclusion of seal at the fold.

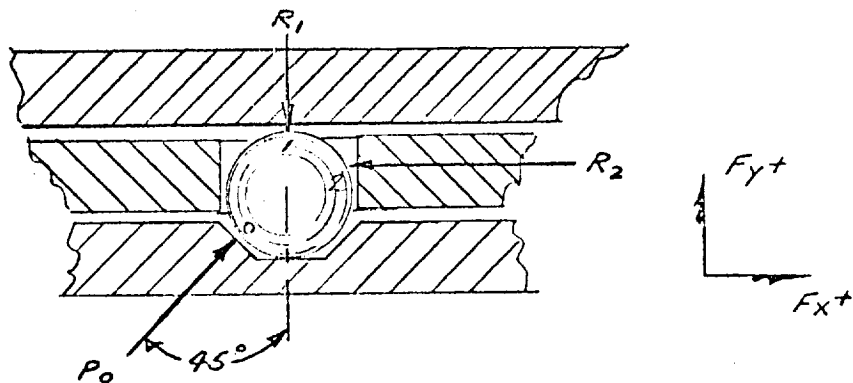
2) Ball Lock

$$\text{Seal area} = \pi r^2 = 6.6 \times 10^2 \text{ in}^2$$

$$\text{Blow apart load} = PA = 6.6 \times 10^3 \text{ lb.}$$

Assuming 16 each 1/2 inch diameter balls and 50% locking efficiency.

$$L/B = \frac{6.6 \times 10^3}{8} = 8.25 \times 10^2 \text{ lb} = \text{load/ball}$$



The spherical ball shall be analyzed for the maximum load 8.25×10^2 lbs. This load is assumed to act on the point (0, 1, 2) as shown.

Assumptions:

The balls and contact surfaces are made to commercial tolerance.

A 50% loading efficiency is assumed

$$N_e = 8$$

$$P_o = \frac{P_h}{N_e \cos 45^\circ} = 1.46 \times 10^2$$

Solving for the reactions R_1 and R_2

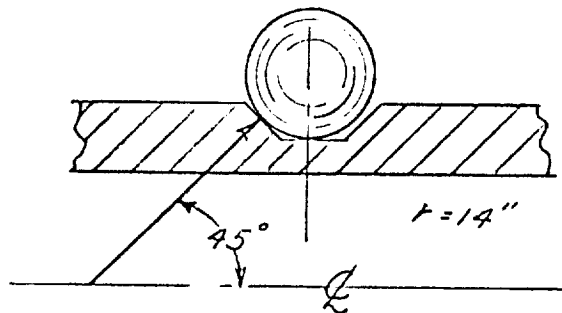
$$\Sigma F_y = -R_1 + P_o \cos 45^\circ = 0$$

$$R_1 = 1.03 \times 10^2$$

$$\Sigma F_x = -R_2 + P_o \cos 45^\circ = 1.03 \times 10^2$$

The contact stresses are evaluated from the Hertz equation.

Geometrical relationship -



Radii of curvature -

$$R_1 = \infty$$

$$R_2 = 20.6$$

Using steel balls and races -

$$E_1 = E_2 = 3 \times 10^7 \text{ psi} \quad \delta_1 = \delta_2 = 3 \times 10^{-1}$$

$$K = \frac{8 E_1^2}{3 [E_2 (1 - \delta_1^2) + E_1 (1 - \delta_2^2)]} = 4.4 \times 10^7$$

$$\delta = \frac{4}{\frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_1'} + \frac{1}{R_2'}} = .495$$

The angle of contact ϕ between the bodies is -

$$\phi = \cos^{-1} \frac{1}{4} \delta \sqrt{\left(\frac{1}{R_1} - \frac{1}{R_1'}\right)^2 + \left(\frac{1}{R_2} - \frac{1}{R_2'}\right)^2 + 2 \left(\frac{1}{R_1} - \frac{1}{R_1'}\right) \left(\frac{1}{R_2} - \frac{1}{R_2'}\right) \cos 2\phi}$$

$$\phi = \cos^{-1} \frac{1}{4} (.495)(.048) = \cos^{-1} 5.95 \times 10^{-3}$$

$$\phi = 1.565 \text{ radians} = 90^\circ$$

$$\omega = 1.00$$

$$\rho = 1.00$$

$$\alpha = 2.00$$

The contact area is circular with a radius given by c , where

$$c = \omega \sqrt[3]{\frac{P\delta}{K}} = 1.18 \times 10^2$$

The maximum compressive stress is S_c , and is given by -

$$S_c = \frac{1.5p}{\pi S_c^2} = 5.01 \times 10^5$$

Which is below the maximum allowed Hertz stress.

The combined deformation at the contact along the axis of load is y -

$$y = \alpha \sqrt[3]{\frac{p^2}{K^2\delta}} = 5.65 \times 10^{-6} \text{ in.}$$

2. HINGE JOINTS: HUB-TO-SPOKE, SPOKE-TO-MODULE

a. General Description

Upon the termination of the station erection sequence the telescoping spoke section, which was initially aligned along the hub axis, has rotated 90° and mated with the extended portion of the hub below the station skirt. The base point design configuration requires that at the final portion of this maneuver, latch pins be extended to capture the mating halves, and tensile actuation of these latches cause the section halves to contact, and further provides seal squeeze. The sealing arrangement stipulated a bellows loaded seal in order to balance the blow apart force.

Two operational problems arise from this design.

The station must be de-erected in order to replace the seal element, since the actuators supply seal squeeze.

A system for compressing the bellows to replace the seal must be provided.

It is noted that if a bellows is used, the actuators cause mating of parts while the seal squeeze is provided by spring action of the bellows.

Therefore, the suggested design modification was

accomplished using the following criteria:

The actuators which are used to clamp the sections together provide seal squeeze to a temporary lip seals only.

Permanent sealing is provided by the manually made-up sealing ring, six-clamp arrangement.

Six clamps were provided as a compromise between erection problems and flange size increase.

In the analysis no attempt was made to define sizes and all loads were assumed to be carried by the connection.

Stress calculations were accomplished to yield representative section parameters.

The design (Figure 5-3) employs an actuator which allows a maximum envelope of contact, in that the envelope is defined by the maximum actuator stroke, the free length of the lug, and the maximum slot width. The actuator is a rotary type operating into a seated slot which is open on one side to receive the lug during the first 180° of lug rotation. The lug is initially aligned normal to the latch plane. Rotation causes the lug to rotate 180° until it contacts the side of the receptacle which is normal to the station surface. Continued rotation causes linear

travel of the lug and screw until the clamping maneuver is completed.

The actuator has a maximum tensile force capacity of approximately three times the blow apart load, to aid in overcoming frictional problems resulting from the erection sequence. Any greater loads would necessitate strengthening the spoke skin which does not seem warranted.

A lip seal is provided to perform a temporary sealing function, with the seal seating force being supplied by a combination of spoke pressurization and external actuator force.

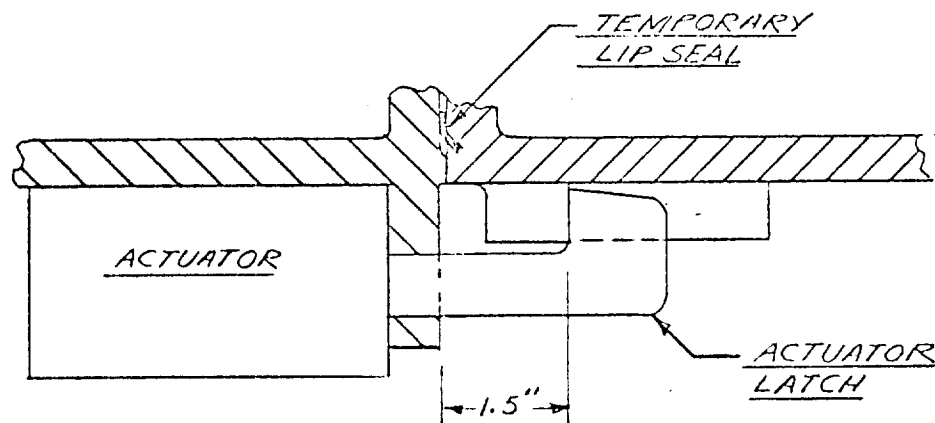
It should be noted that the screw type actuator can be very reliable since the lubrication can be amply applied and will last during erection, and further, upon full travel, no further power is required since backdriving cannot occur.

The permanent seal is made-up manually by the astronaut crew and comprises a complete seal ring carrying two 'O' ring type seals. Six equally spaced, bolted clamps are required to provide seal squeeze. This design allows for simple replacement of seals without total depressurization. Limited leakage would occur through the lip seal.

The same clamping arrangement is used for the spoke-module hinge joint.

b. Representative Stress and Sizing Calculations

1) External Clamp



The load on the actuator will consist of the blow apart load, the seal load and a factor included to allow the clamp to close up any small gap remaining at the joint during station erection. The load resulting from the lip seal is small and is included in the factor to close the gap.

Shell Diameter = 56 inches

Blow Apart Pressure = 10 psi

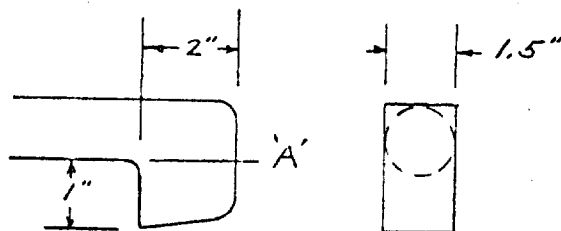
$$\text{Area} = 2462 \text{ in}^2$$

$$\text{Blow Apart Load} = 24,620 \text{ lbs.}$$

The maximum load is increased by a factor of two to take into account the extra forces imposed by binding during erection. No higher factor is practical since the skin will have to be reinforced locally to take this higher load.

$$\text{Total Load} = 49,240 \text{ lbs}$$

The clamp material will be steel to minimize deflections and increase tensile stress allowable in the rod.



$$I(A) = \frac{bh^3}{12} = 1 \text{ in}^4$$

Bending stress at 'A', beam considered as a compromise of end loaded and uniformly loaded cantilever.

$$S_b = \frac{Mc}{I} = 18,465 \text{ psi}$$

$$S(A) = 16,410 \text{ psi}$$

$$S(\text{total}) = 34,875 \text{ psi}$$

$$\text{Deflection at 'A'} = y = \frac{.49Wl^2}{EI} = .001 \text{ in.}$$

$$\text{Rod Diameter} = 1.5 \text{ in.}$$

$$\text{Area Rod} = 1.767 \text{ in}^2 \quad I (\text{Rod}) = .248 \text{ in}^4$$

$$\text{Tensile Stress in Rod} = 27,860 \text{ psi}$$

$$\text{Assume Rod Length} = 4 \text{ in.}$$

Deformation of rod due to direct tension load y is given by,

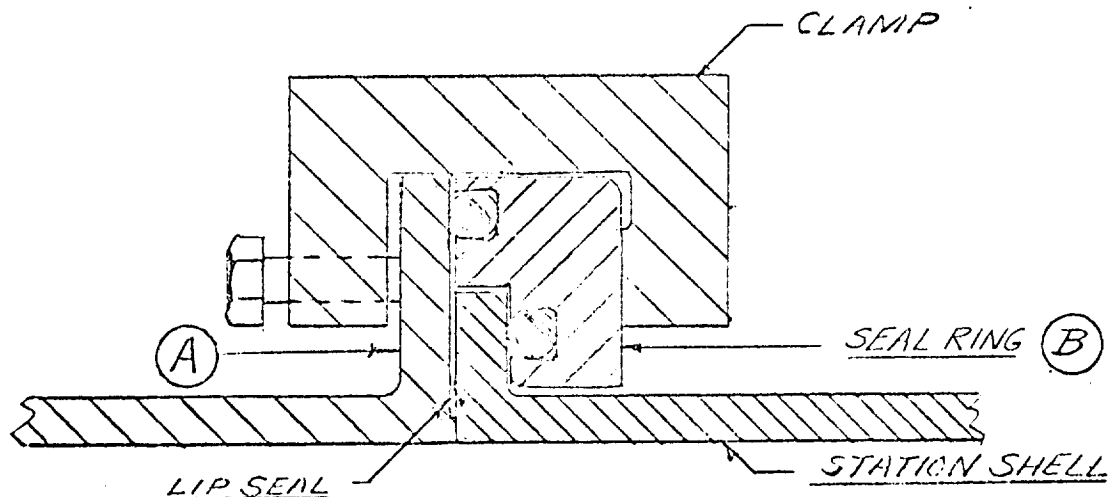
$$y = \frac{Pl}{AE} = .004 \text{ in.}$$

$$\text{Moment at 'A'} = 75 Wl = 36,930 \text{ in-lbs}$$

$$\text{Deflection at 1 in.} = .007 \text{ in.}$$

$$\text{Total Deflection Actuator Clamp Arm} = .012 \text{ in.}$$

2) Internal Clamp and Seal Ring



The loading on this joint consists of the seal flange and the blow apart loads.

The clamp must take both loads while flange (A) and ring (B) must take only the seal pressure load between clamps.

Assume .500 inch diameter, 50 durometer 'O' rings.

Use sealing pressure of 125 psi with total allowable deflection 10% diameter = .050 inch.

From Figure 4-4, flange load = 24 lbs/inch

For two 'O' rings = 48 lbs/inch

Circumference = 145 inches

Total Seal Load = 6960 lbs.

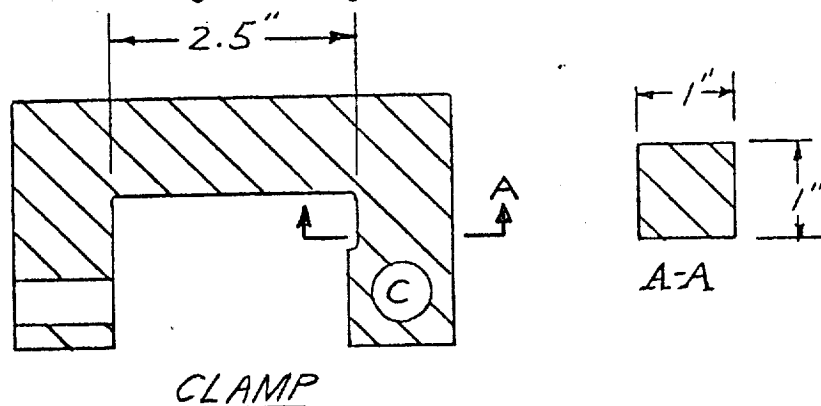
Area subjected to 10 psi blow apart load -

$$= \pi r^2 = 1800 \text{ in}^2$$

Blow Apart Load = 18,000 lbs.

Using six clamps -

$$\frac{6960}{6} + \frac{18,000}{6} = 4160 \text{ lbs/clamp}$$



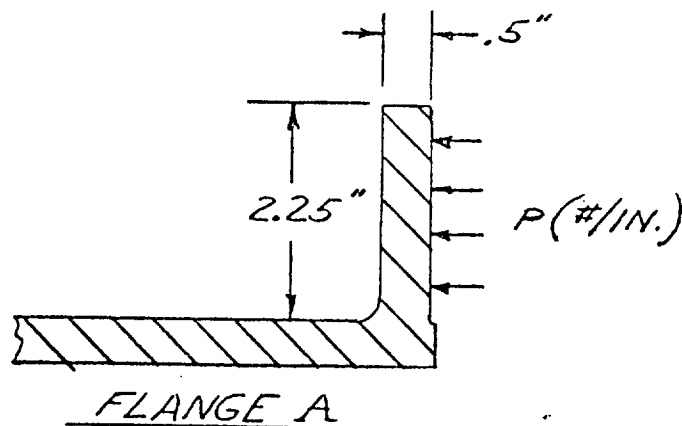
Tension deformation on clamp y_1

$$y_1 = \frac{pl}{AE} = .001 \text{ in.}$$

Consider leg (C) a cantilever with a loading between uniform and concentrated so deflection deformation on clamp = y_2 .

$$y_2 = .29 \frac{wl^3}{EI} = .008 \text{ inches}$$

Total Deformation of Clamp = $y_1 + y_2 = .009 \text{ in.}$



With six clamps, distance between clamps

$$= \frac{145}{6} = 24.1 \text{ inch}$$

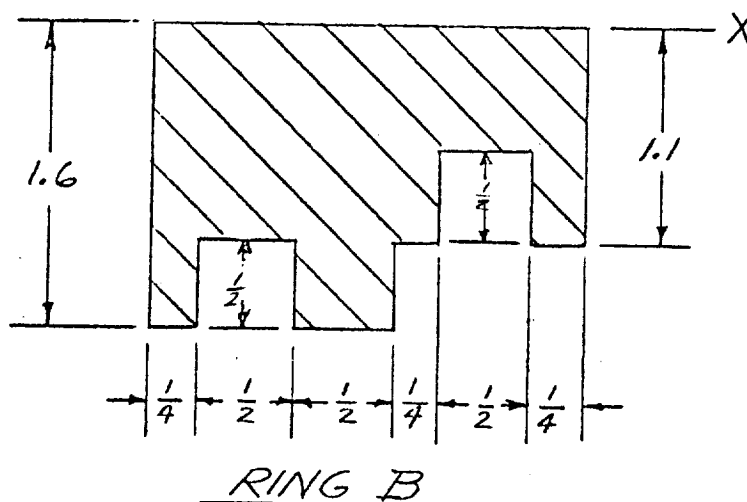
Since ratio b/a is small, the flange may be treated as a uniformly loaded strip of unit width with one end built in and other end free.

$$\text{Max Deflection} = W = \frac{.125 pb^4}{D}$$

$$D = \text{Flexural Rigidity} = \frac{Eh^3}{12(1-\delta^2)}$$

$$p = \frac{48}{2.25} = 21.3 \text{ lbs/in.}$$

$$W = .0005 \text{ in. (Negligible)}$$



$$\bar{y} = \frac{\int y da}{A} = .636 \text{ in.}$$

$$I_x = 1.497 \text{ in}^4$$

$$I_{xg} = I_x - A (\bar{y})^2 = .447 \text{ in}^4$$

Consider the ring between clamps as a compromise between a simply supported and fixed end beam and setting the maximum allowable deflection equal to .040 in. (.009 in. is possible in clamp)

l = max. allowable span between clamps

l = 26.3 in.

This clamp design will be used in the internal portion of the hinge joints and the inter-module airlock

3. ROTATING JOINT SEAL

a. General Description

Figure 5-9 shows the conceptual arrangement of the rotating seal and the associated components. In this concept, two annular rings are mounted adjacent to the bearings.

The primary purpose of these rings are to retain the bearings and the seals.

During the launch cycle, the clamping ring is not

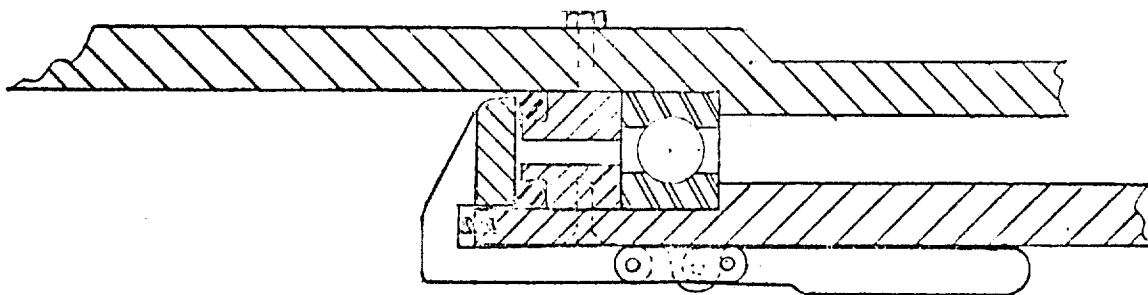
in contact with the seals, thus allowing friction-free rotation. Six quick-acting clamps are located peripherally around the clamping ring. After the station is erected, personnel entering this area can manually actuate the over-center clamps, thus closing off the area between the seals and providing the proper seal squeeze. The following advantages are offered by this design.

Seals may be replaced from within the spoke to module airlock without disrupting normal station functions.

Friction-free rotation of the joint is attained during station erection.

The 'O' ring seals are not subjected to abrasion.

b. Representative Stress and Sizing Calculations



The location of the clamp ring causes the seal seating stress to remain constant when the joint is pressurized. The analysis is accomplished utilizing six clamps as a trade off between optional flange sizing and erection procedure complications. Additional clamps would obviously decrease the flange size and weight.

Assumptions:

Section Diameter = .500 in.

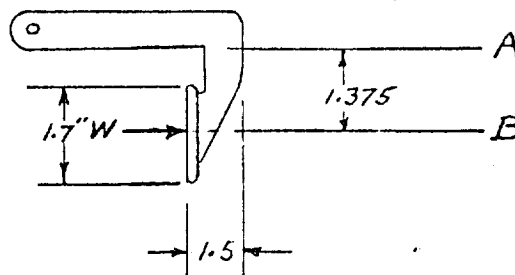
Durometer = Shore A 50

Seating Pressure = 125 psi

From Figure 4-4

Flange Load = 24 lb/in for each 'O' ring

Allowable Deflection = 10% = .050 in.



Material: 2024-T4 Aluminum

Assuming six clamps, 2 in. wide x .5 in. deep (section A)

Sealing Circumference = 175 in.

Total Load = 8400 lbs.

$$\text{Clamp Load} = 8400/6 = 1400 \text{ lb/clamp}$$

Checking as an end loaded cantilever beam -

$$I_{(A)} = \frac{bh^3}{12} = .02 \text{ in}^4$$

$$s_{(A)} = \frac{Mc}{I} = 24,050 \text{ psi}$$

$$y_{(A)} = \frac{Wl^3}{EI} = .006 \text{ in.}$$

This reduces the allowable deflection of the flange ring between clamp points to .044 in.

4. SPOKE SLIDING JOINT

a. General Description

Figure 5-16 shows a conceptual arrangement of the spoke sliding seal and its associated components. Due to the extreme length of travel (approximately 153 inches) it is obvious that the seal must be made-up at the end of travel. In this concept, a preformed L shaped elastomer seal is secured to the end of the extendable spoke by means of an annular clamping ring. The expanding clamp (reference Figure 5-17) is collapsed and stored on suitable hangers during launch. After the telescoping section is extended to the

desired length, the clamp is removed from stowage, and positioned on the inner surface of the seal by station personnel. The clamp may then be expanded into a full ring inside the seal by operating the over-center levers. The required seal squeeze is obtained by adjusting a turnbuckle arrangement. The following advantages are realized by the use of this design.

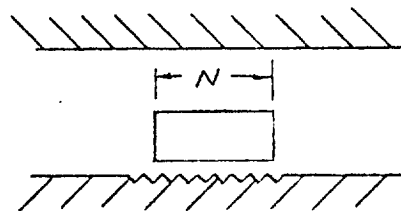
The seal is not required to slide 153 inches in the squeezed condition, thus reducing the power required to erect the station.

The amount of squeeze is readily adjustable. Concentricity requirements of the structural members are not as stringent as with a piston-type seal.

The seal may be easily replaced.

b. Representative Stress and Sizing Calculations

Sliding Spoke Joint (Expanding Clamp)



Required Sealing Pressure = 125 psi

Maximum Allowable Deflection = .050 in.

b = Effective Sealing Width

b_o = Basic Gasket Seating Width

b_o = 7/16 N = 7/16 in.

$$b = \sqrt{\frac{b_o}{2}} = .33 \text{ in.}$$

Assuming 3 clamps -

Length per Segment = 58 in.

Load per Segment = 2392 lbs.

I_{req'd} = 7.2 in⁴

5. INTER-MODULE AIRLOCK

a. General Description

This design concept adheres to the physical parameters and the erection sequences described in Figure 3-6, Base Point Design.

Figure 5-10 shows the seal retainer ring secured to one half of the air lock by bolts which serve to hold the ring in place during launch and to allow personnel to replace the defective seals during station without disassembling the airlock.

Assembled, the seal retainer ring forms an integral

part of the tunnel flange. A lip seal serves to form a temporary seal when each end of the tunnel is brought into conjunction during erection. Seal squeeze is provided by an actuator until personnel enter the station to attach the final clamps.

b. The Module Access Door

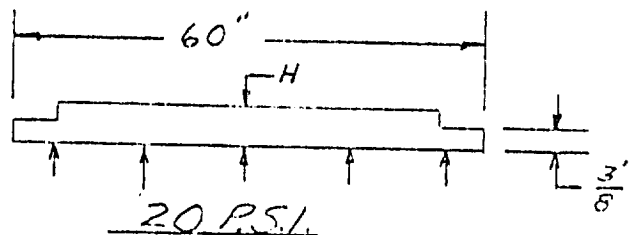
The door and latch design shown in Figures 5-11 through 5-15 is recommended in all areas except the vehicle docking door where additional criteria restrict the door design as explained in previous section. Several alternate approaches are shown. Figure 5-11 illustrates a modification of the base point design incorporating two 'O' ring seals instead of one. The use of two seals is recommended in vacuum applications for the major purpose of decreasing the leakage rate by increasing the permeation path. It also adds a factor of safety against partial or complete loss of one seal. Seals installed as shown, however have disadvantages. The seating area of the seal would be exposed to damage by the crew moving through the doorway. The seals would then slide across this damaged area each time the door is used, leading to their failure. Installation of these rings involves stretching them

over the door diameter and rolling them into the groove increasing the possibility of damage. Figures 5-12 and 5-13 illustrate a type of seal that has been used successfully on vacuum applications and eliminates some of the problems inherent in Figure 5-11. However, no definite leak information is available on this construction. It also appears to be subject to damage when installing the door. From a system and economic standpoint it would also follow that using a standard 'O' ring configuration would be desirable. This leads to the recommended configuration shown in Figure 5-14 and 5-15. The use of 'O' ring type seals eliminates the major drawbacks to the Figure 5-12 and 5-13 design. A minimum of sliding contact is involved when opening and closing the door and the seating flanges are not subjected to damage by the crew movement through the area. The critical sealing problem is to hold the pressure in the module. Under this condition, the pressure holds the door down tightly on the seal. If the opposite condition occurs it is not as critical to maintain a seal as leakproof since the airlock can be evacuated. Installation of the seals in this configuration involves no stretching or rolling.

c. Representative Stress And Sizing Calculations

1) Module Access Door

The door will be considered as a flat plate for this analysis.



When subjected to 20 psi internal pressure during pre-erection, the door can be considered as a plate simply supported on the sealing flanges except in the area of the hinge and latch where moments due to these constraints must be considered.

Treating the door as an edge clamped circular plate -

$$M_c = 2250 \text{ lbs/in.}$$

$$M_d = 556 \text{ lbs/in.}$$

$$M_{\text{total}} = 2800 \text{ lbs/in.}$$

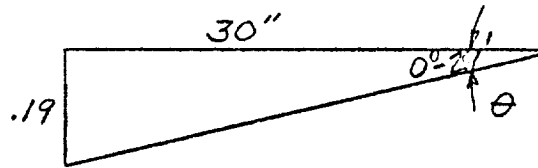
The hinge and latch will provide self support to take these moments. The shear loads are -

$$Q = 300 \text{ lbs/in.}$$

$$S_s = 800 \text{ psi}$$

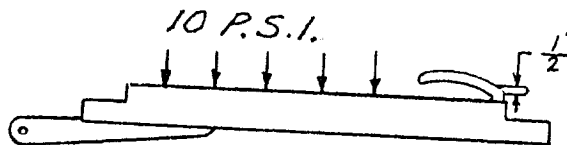
Treating the honeycomb section as equivalent to solid two inch aluminum section, the deflection at the door center equals .19 inches.

The angular deflection, θ , equals $\tan^{-1} .006$.



Therefore, the angular deflection is negligible.

2) Door Latch



When erected the module may be depressurized and the door will be subjected to 10 psi from the reverse direction. With a representative latch section of 2 in. x $\frac{3}{8}$ in. -

$$S_c = 28,200 \text{ psi}$$

This stress is below the double shear allowable steel of 29,400 psi.

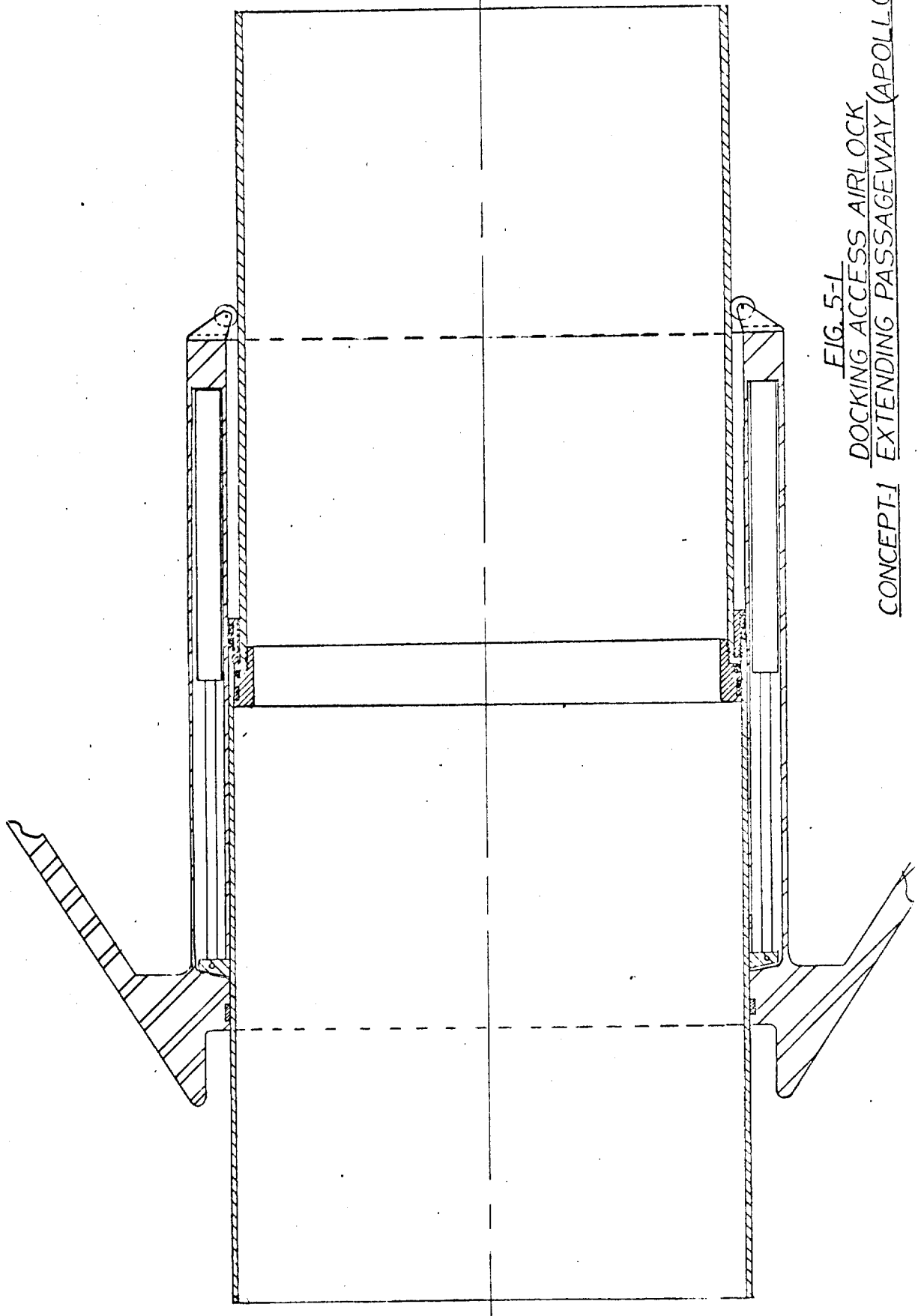


FIG. 5-1
DOCKING ACCESS AIRLOCK
CONCEPT-1 EXTENDING PASSAGEWAY (APOLLO)

E-32

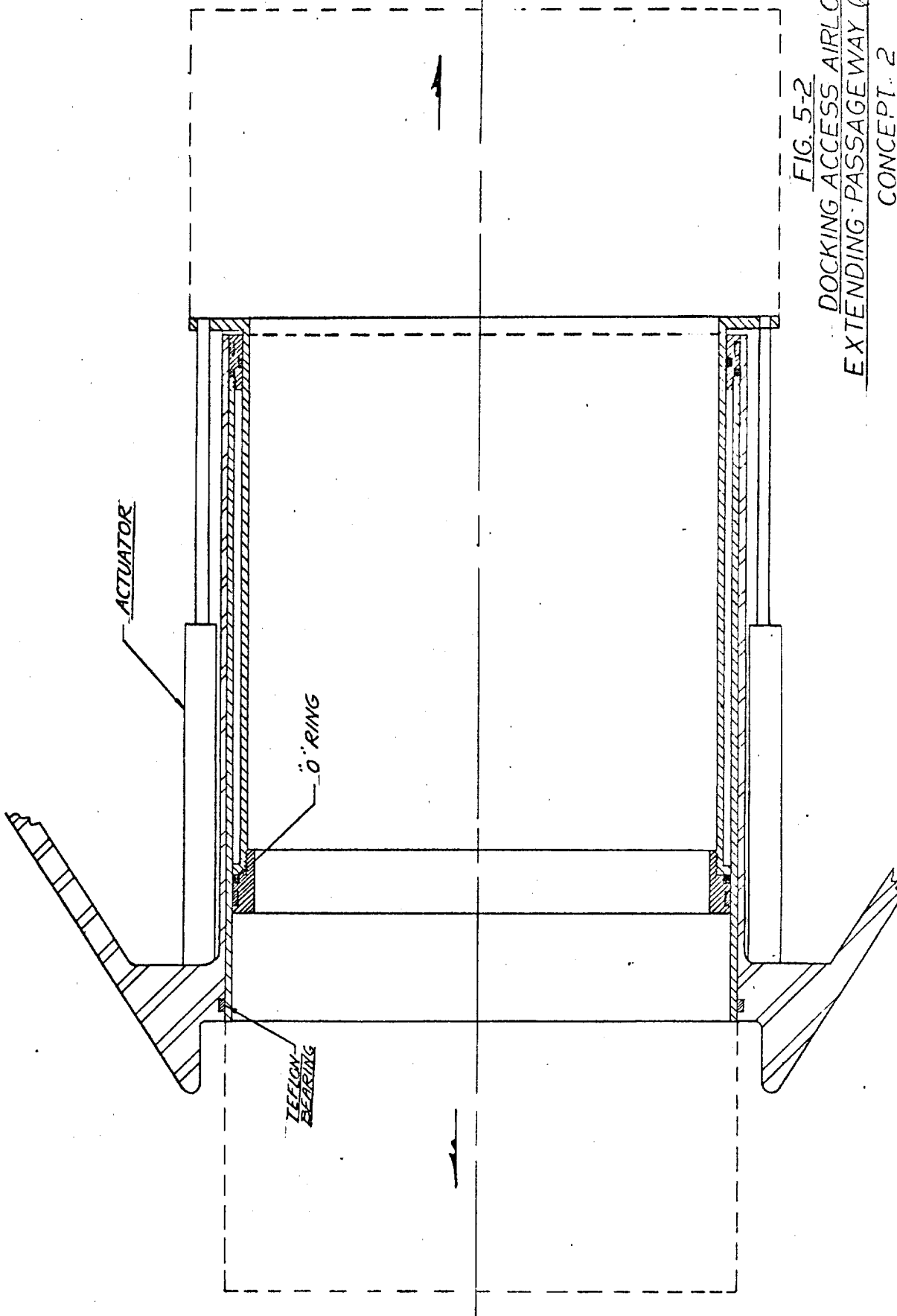


FIG. 5-2
DOCKING ACCESS AIRLOCK
EXTENDING PASSAGE WAY (APOLLO)
CONCEPT 2

E-33

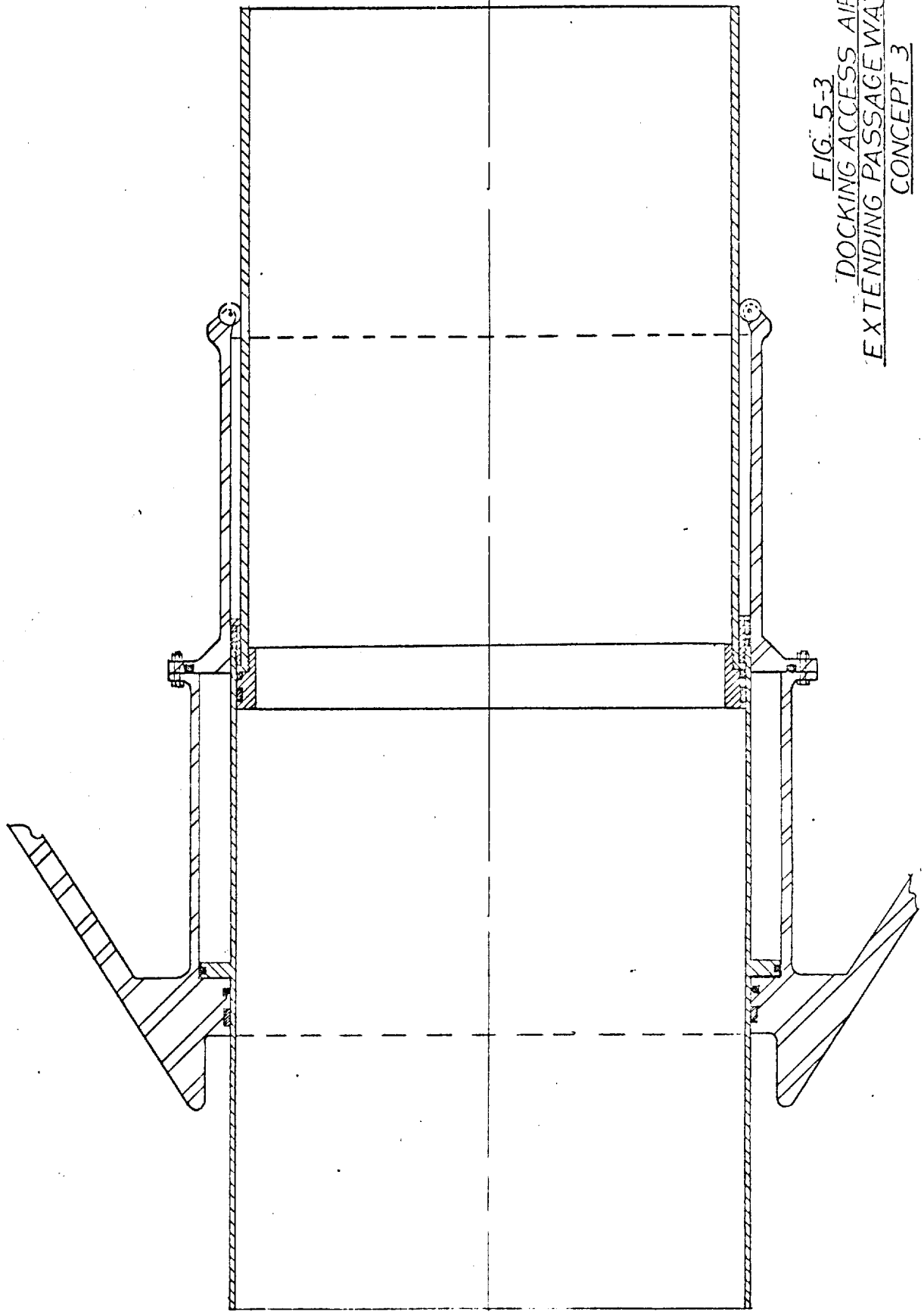


FIG. 5-3
DOCKING ACCESS AIRLOCK
EXTENDING PASSAGE WAY (APOLLO)
CONCEPT 3

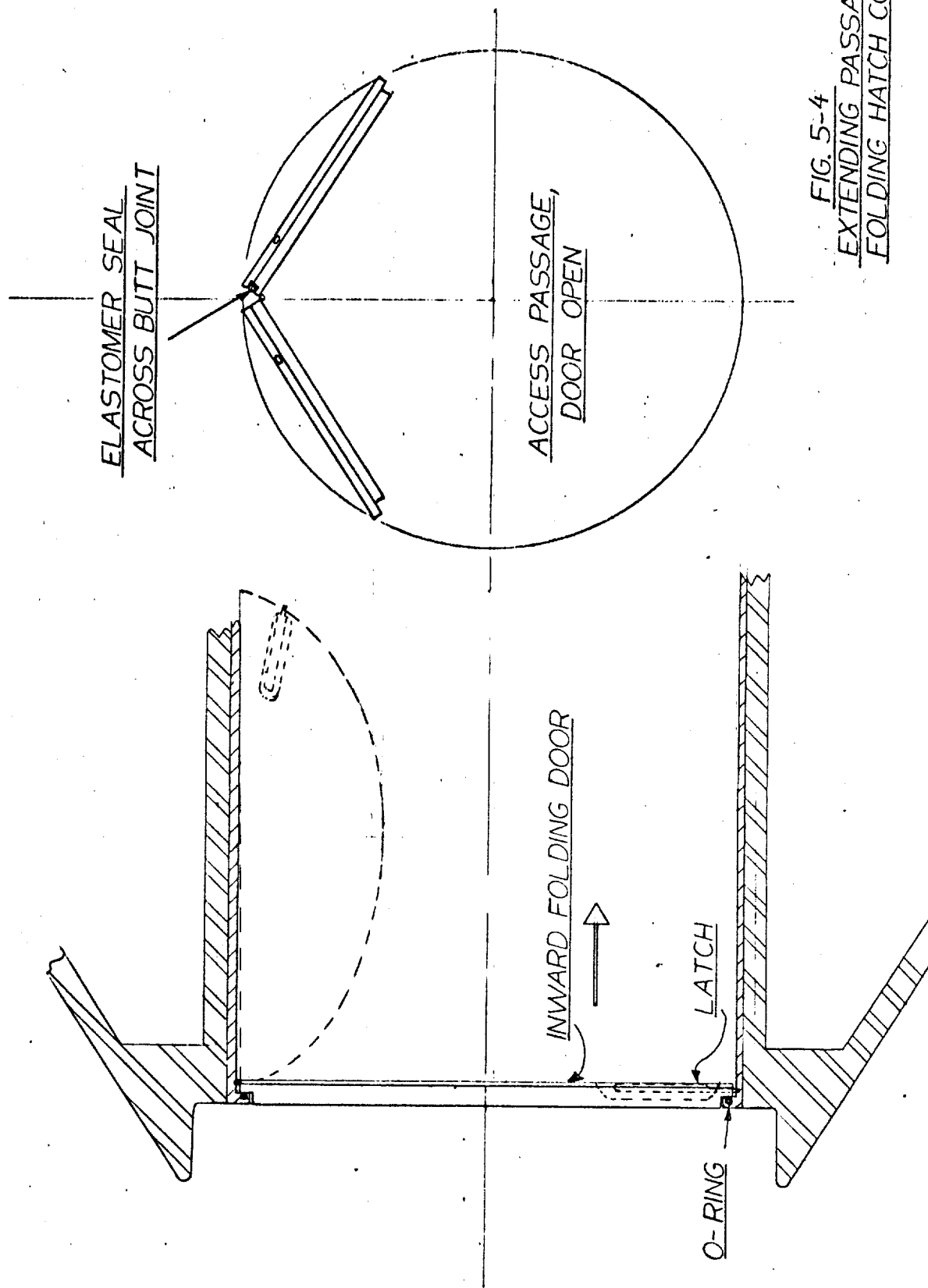


FIG. 5-4
EXTENDING PASSAGEWAY
FOLDING HATCH COVER

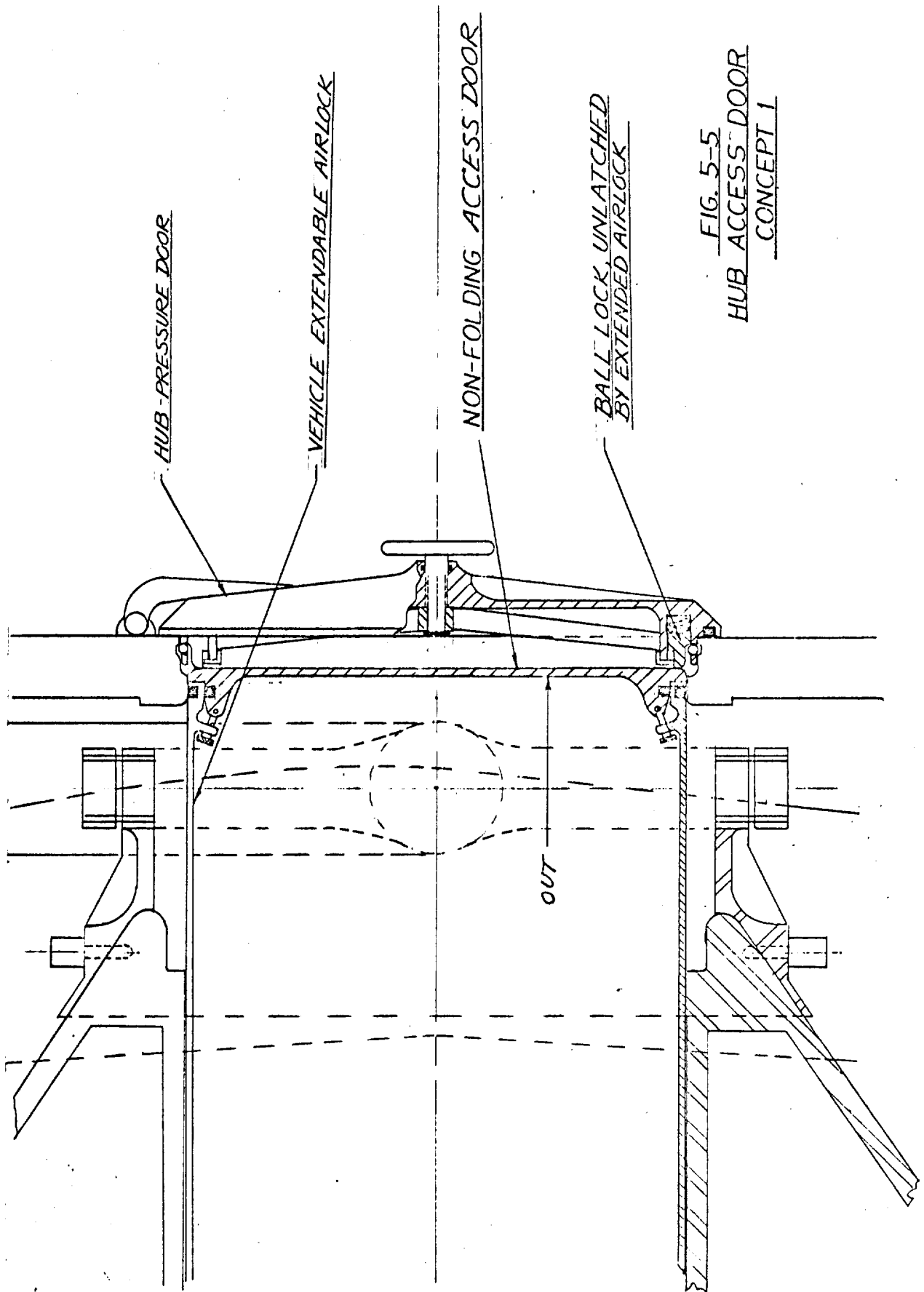


FIG. 5-5
HUB ACCESS DOOR
CONCEPT 1

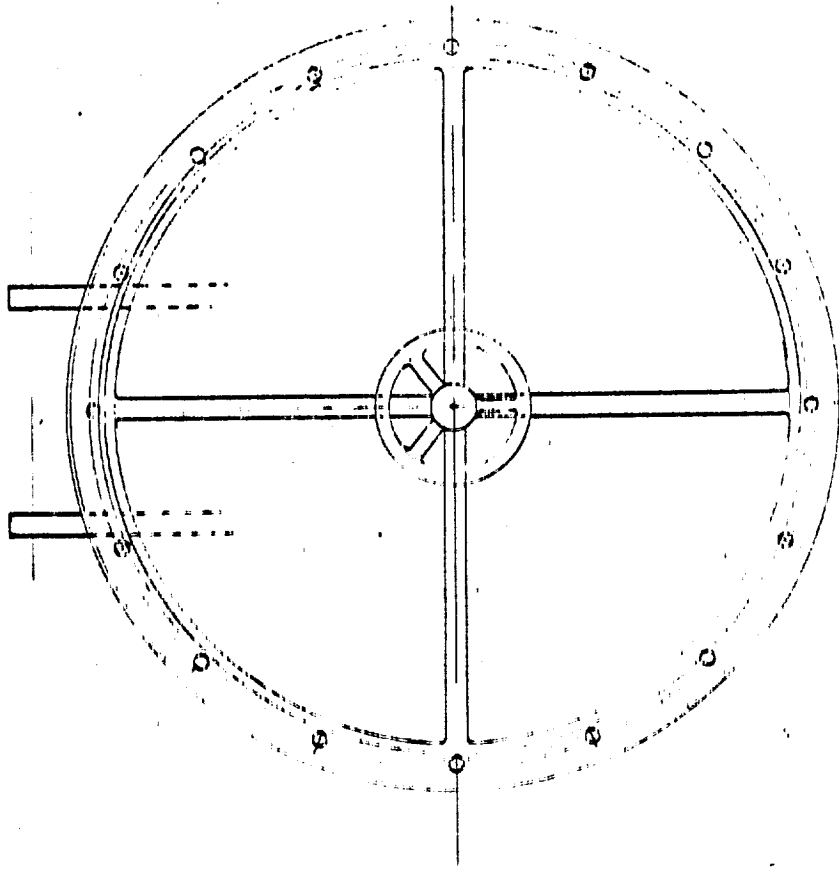
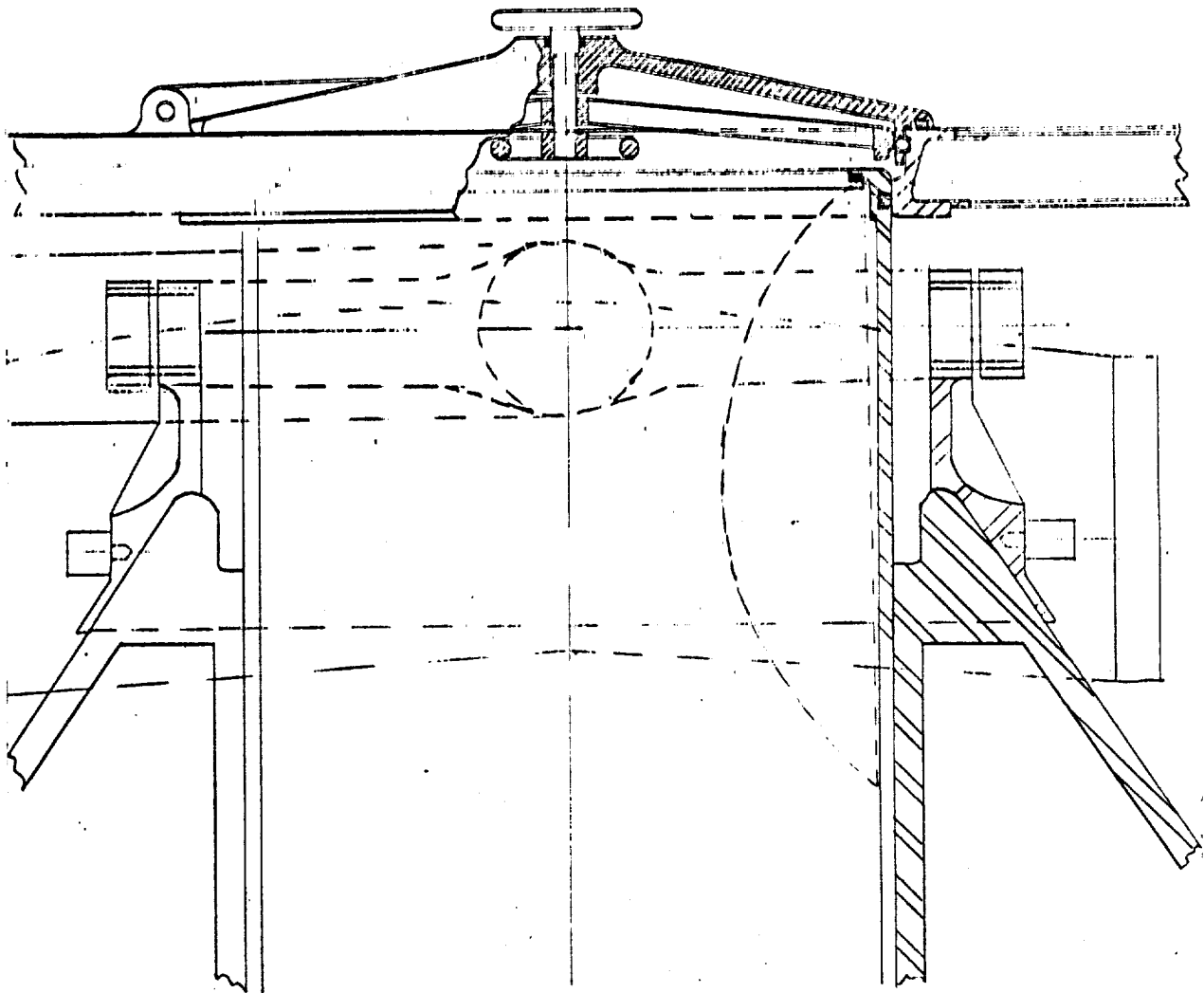


FIG. 5-6
HUB ACCESS DOOR
CONCEPT 2



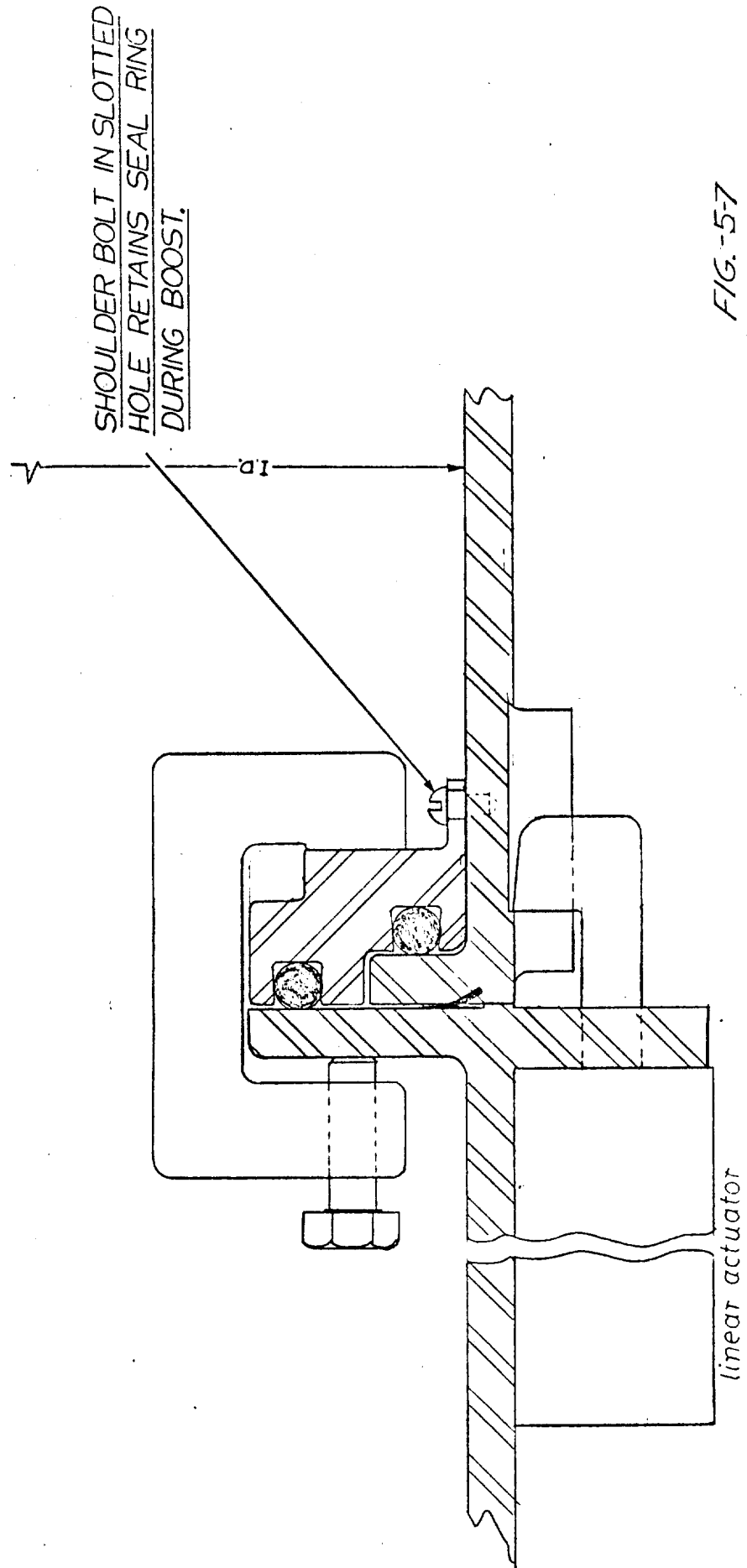
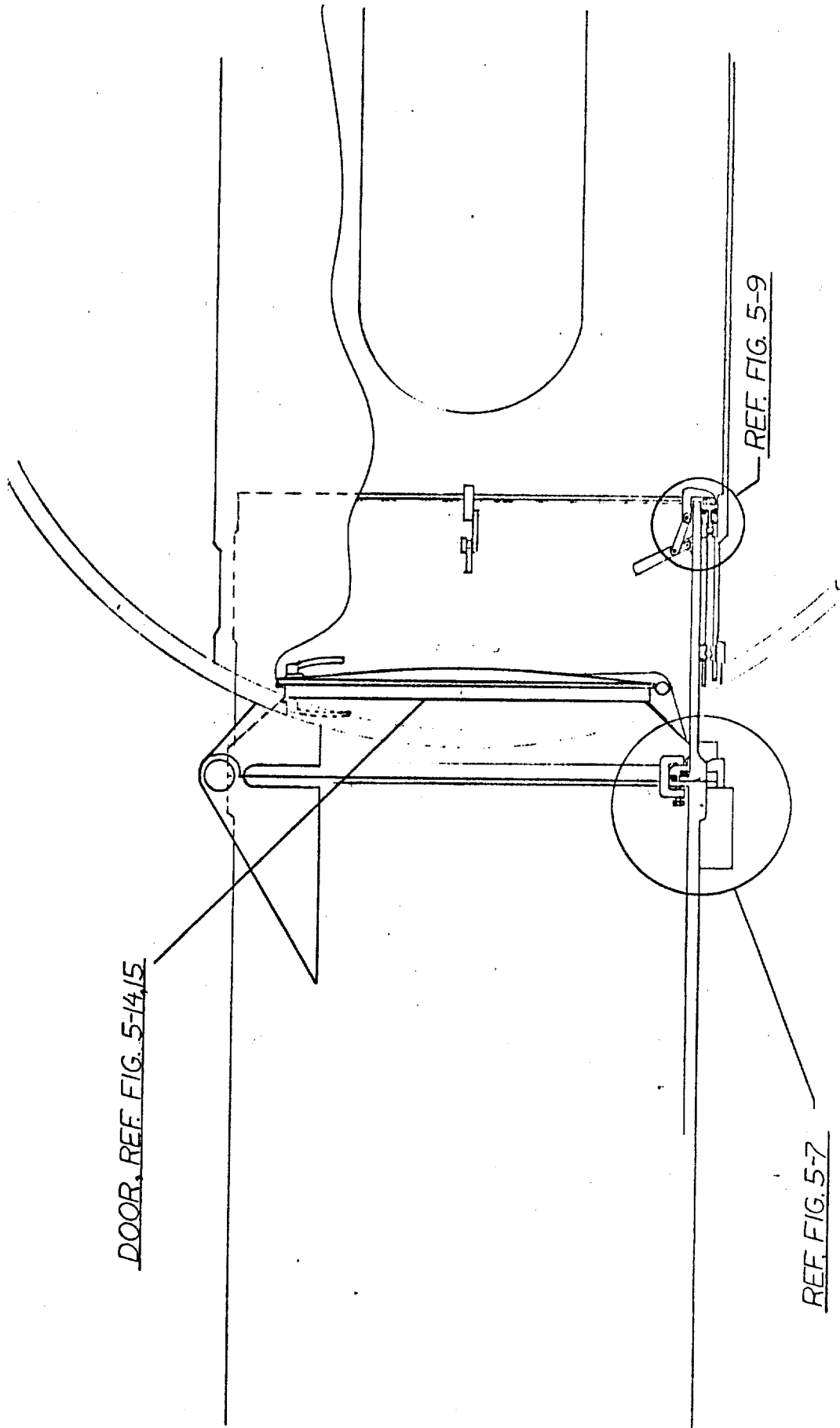


FIG. -5-7
HUB TO SPOKE HINGE JOINT



DOOR, REF. FIG. 5-14, 15

REF. FIG. 5-9

REF. FIG. 5-7

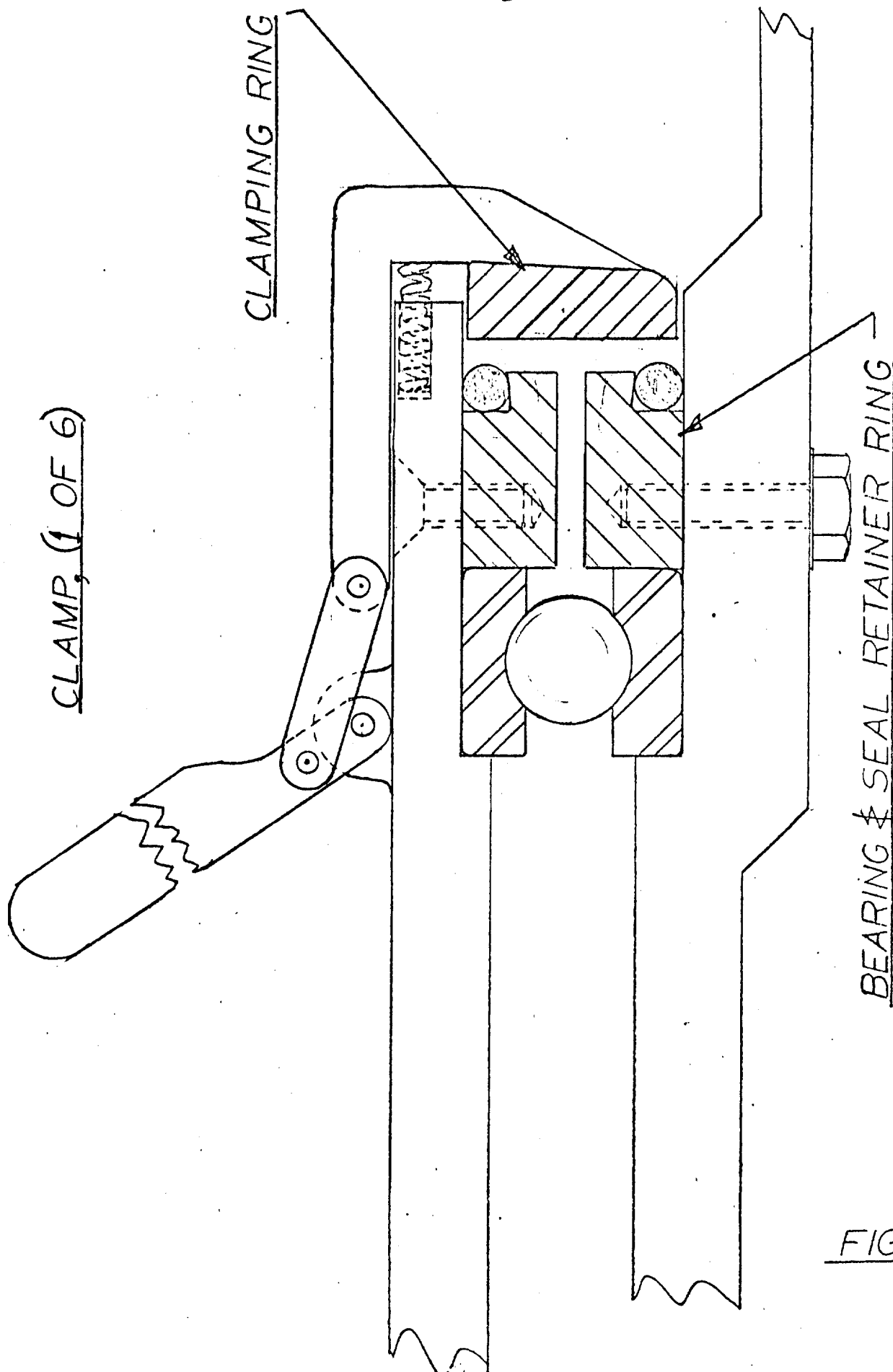
FIG. 5-8
SPOKE TO MODULE HINGE JOINT

CLAMP, (1 OF 6)

E-39

CLAMPING RING

BEARING & SEAL RETAINER RING



ROTATING JOINT SEAL
CONCEPT

FIG. 5-9

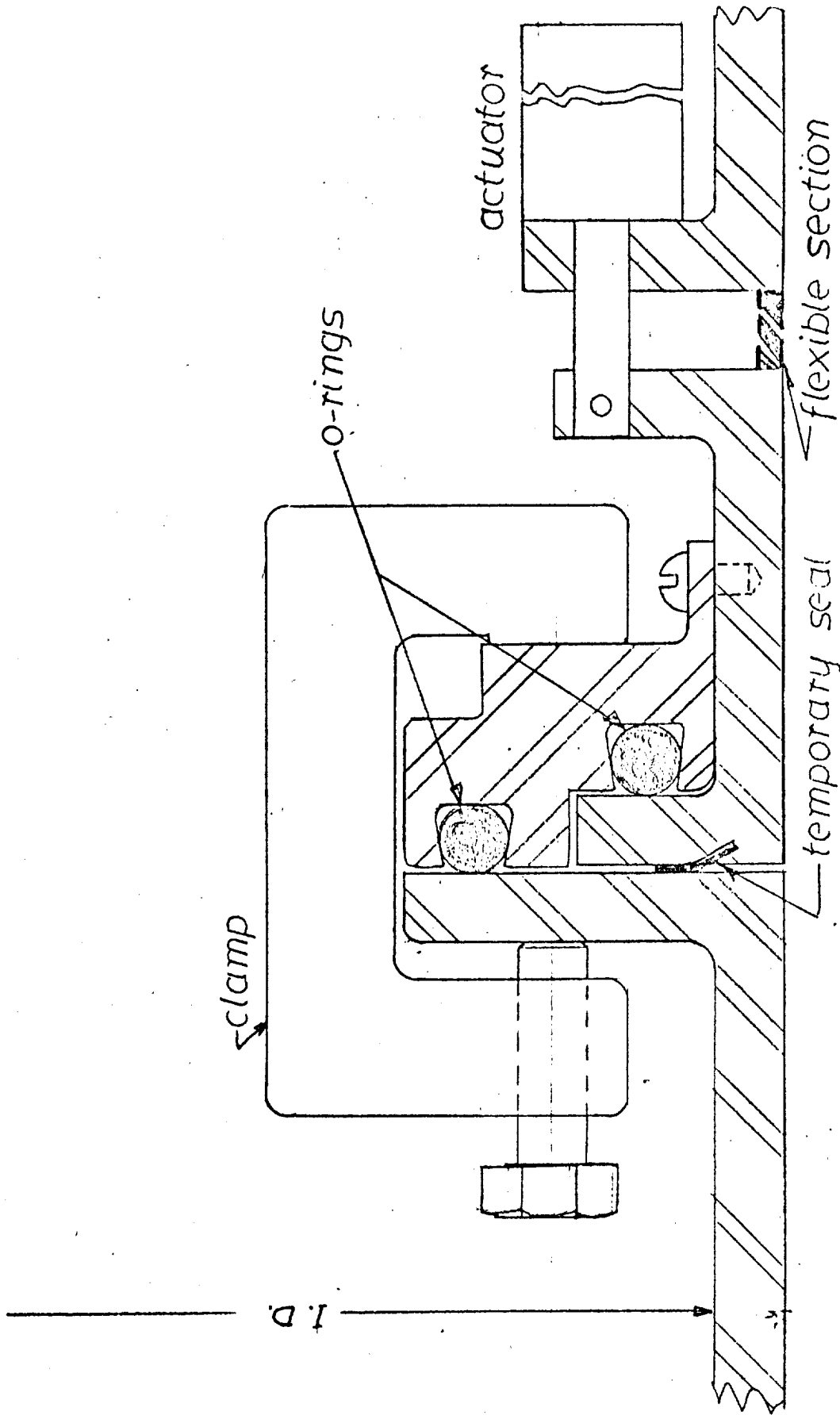


FIG. 5-10
MODULE JOINT INTERFACE AIRLOCK
SEAL & LATCH CONCEPT

E-41

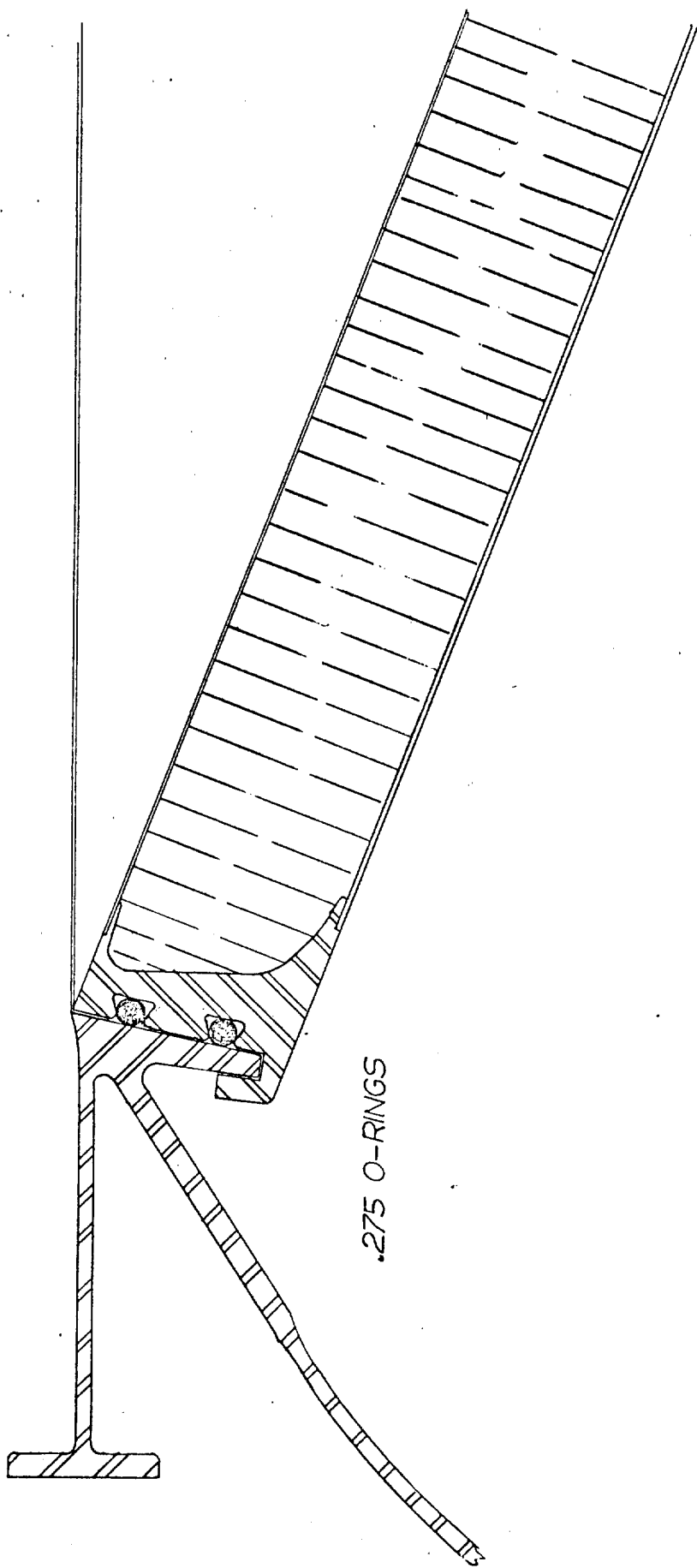


FIG. 5-II
AIRLOCK ACCESS DOOR
CONCEPT 1

F-42

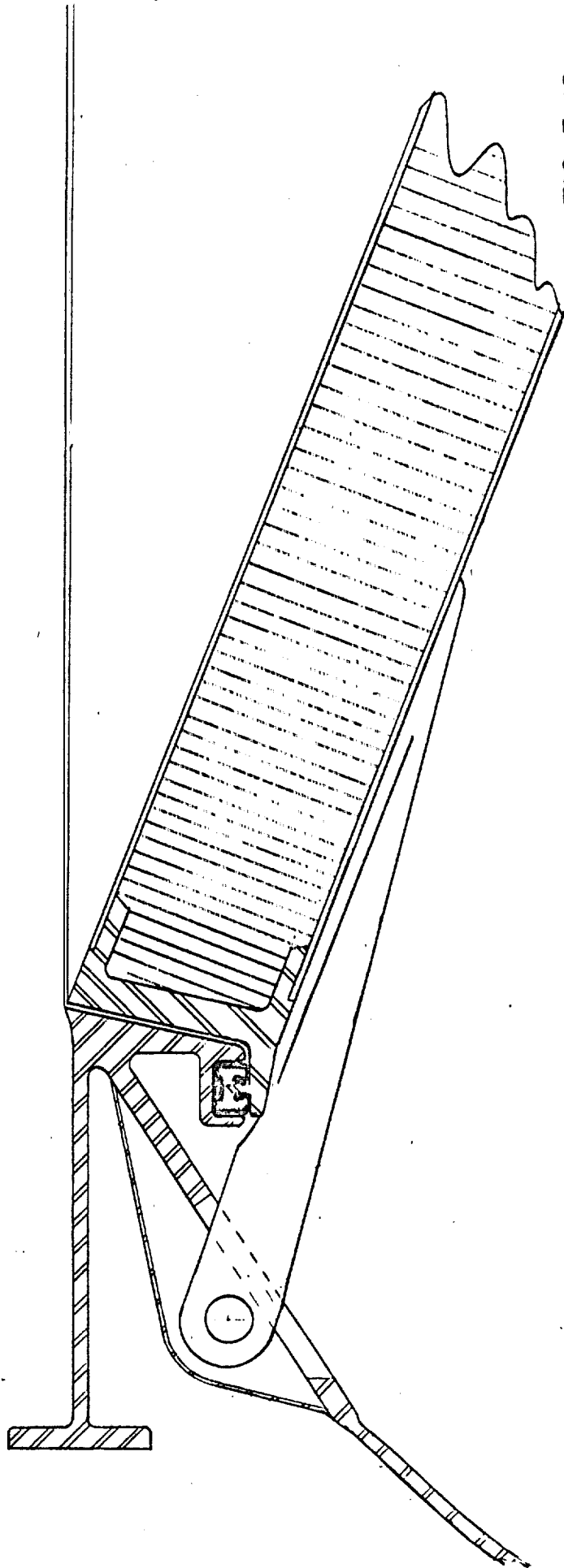


FIG. 5-12
AIRLOCK ACCESS DOOR & LATCH
CONCEPT 2

E-43

CAM SURFACE COMPRESSES
DOOR AND SHAFT SEALS

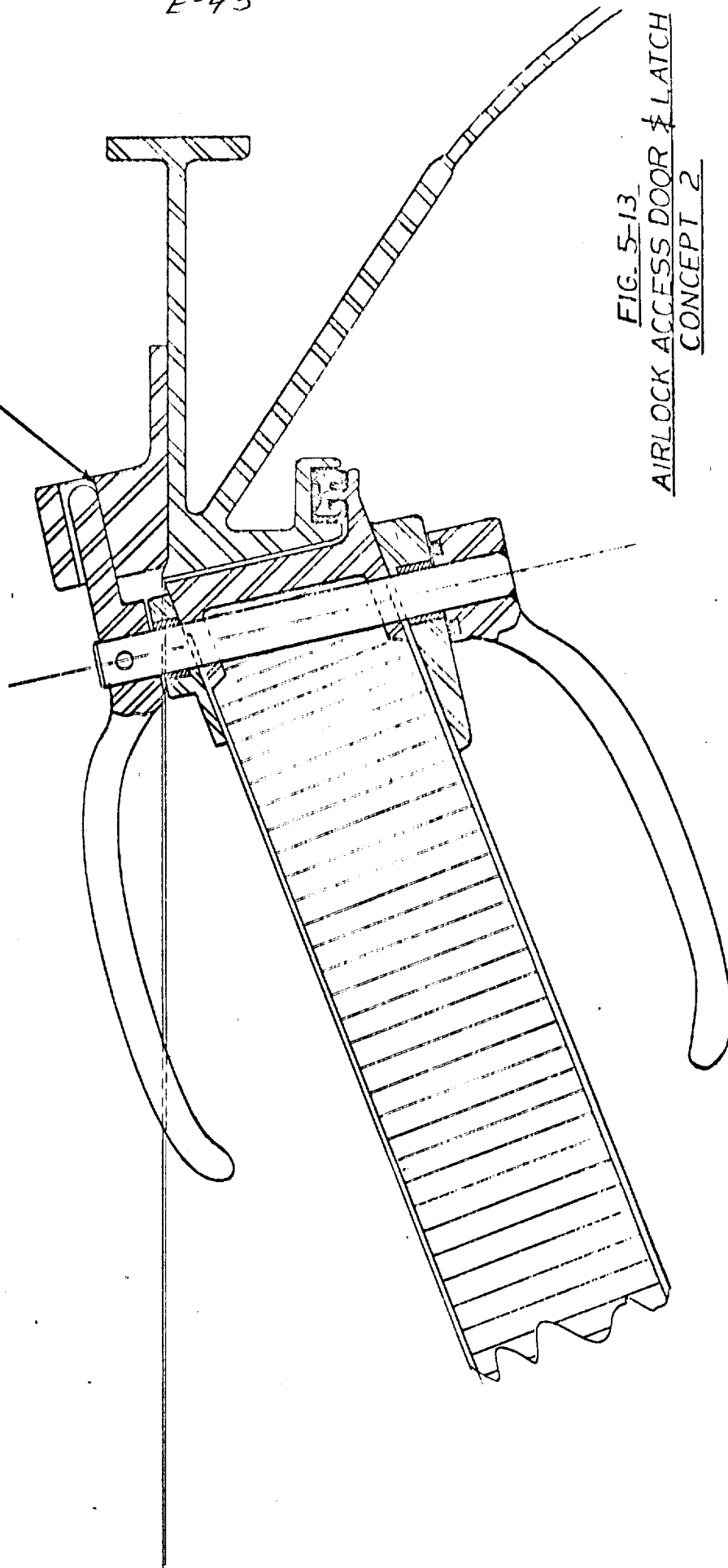


FIG. 5-13
AIRLOCK ACCESS DOOR LATCH
CONCEPT 2

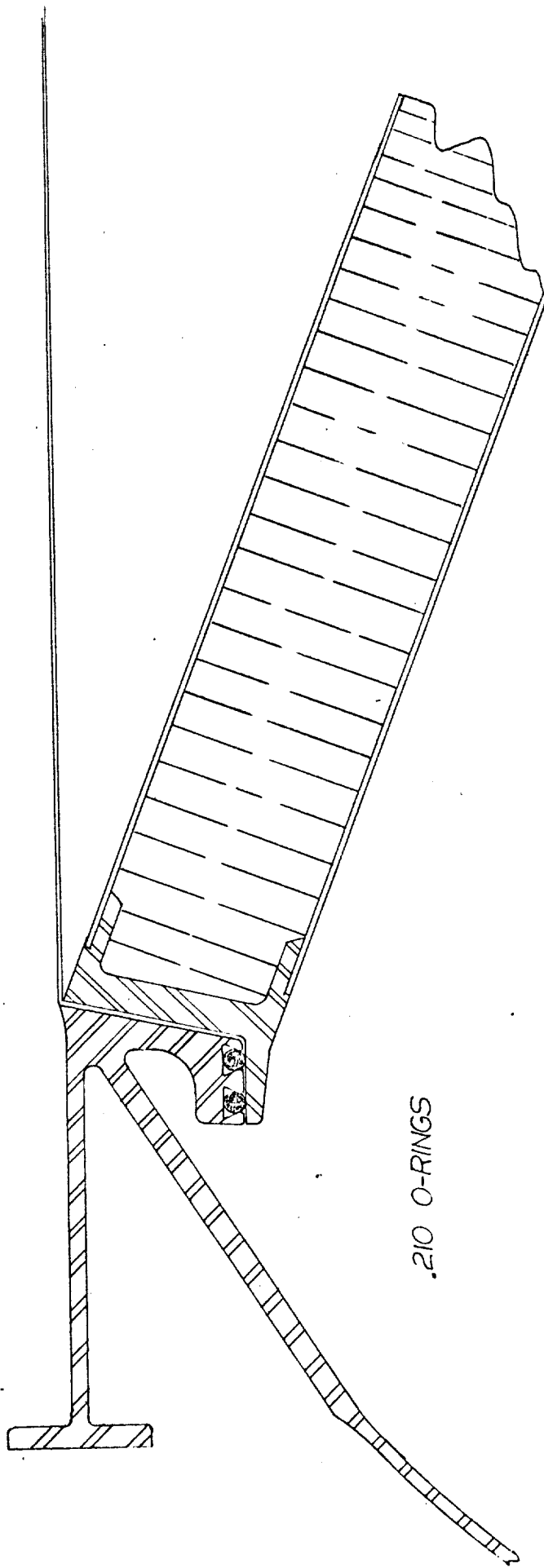


FIG. 5-14
AIRLOCK ACCESS DOOR
CONCEPT 3

210 O-RINGS

E-45

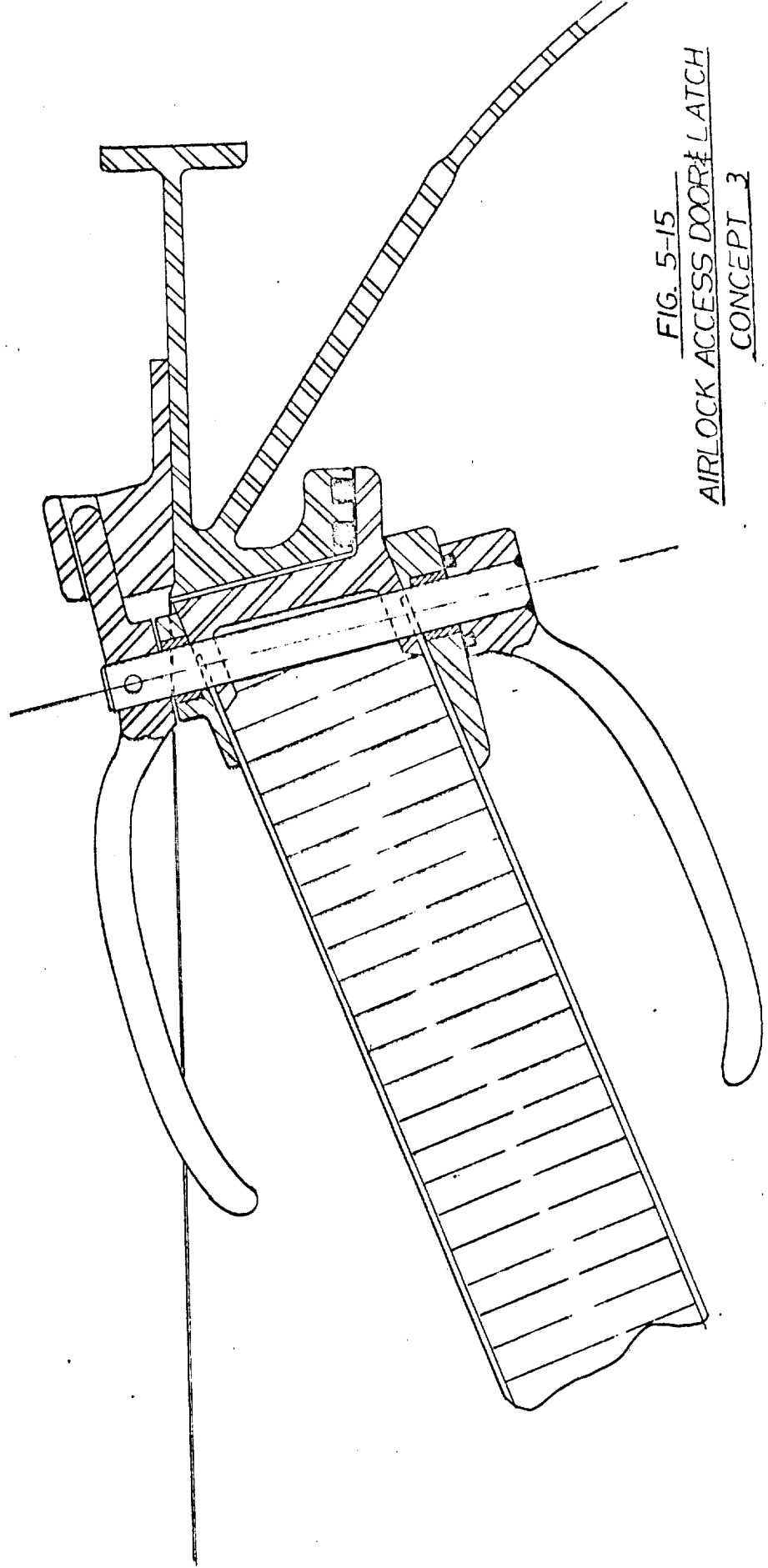


FIG. 5-15
AIRLOCK ACCESS DOOR LATCH
CONCEPT 3

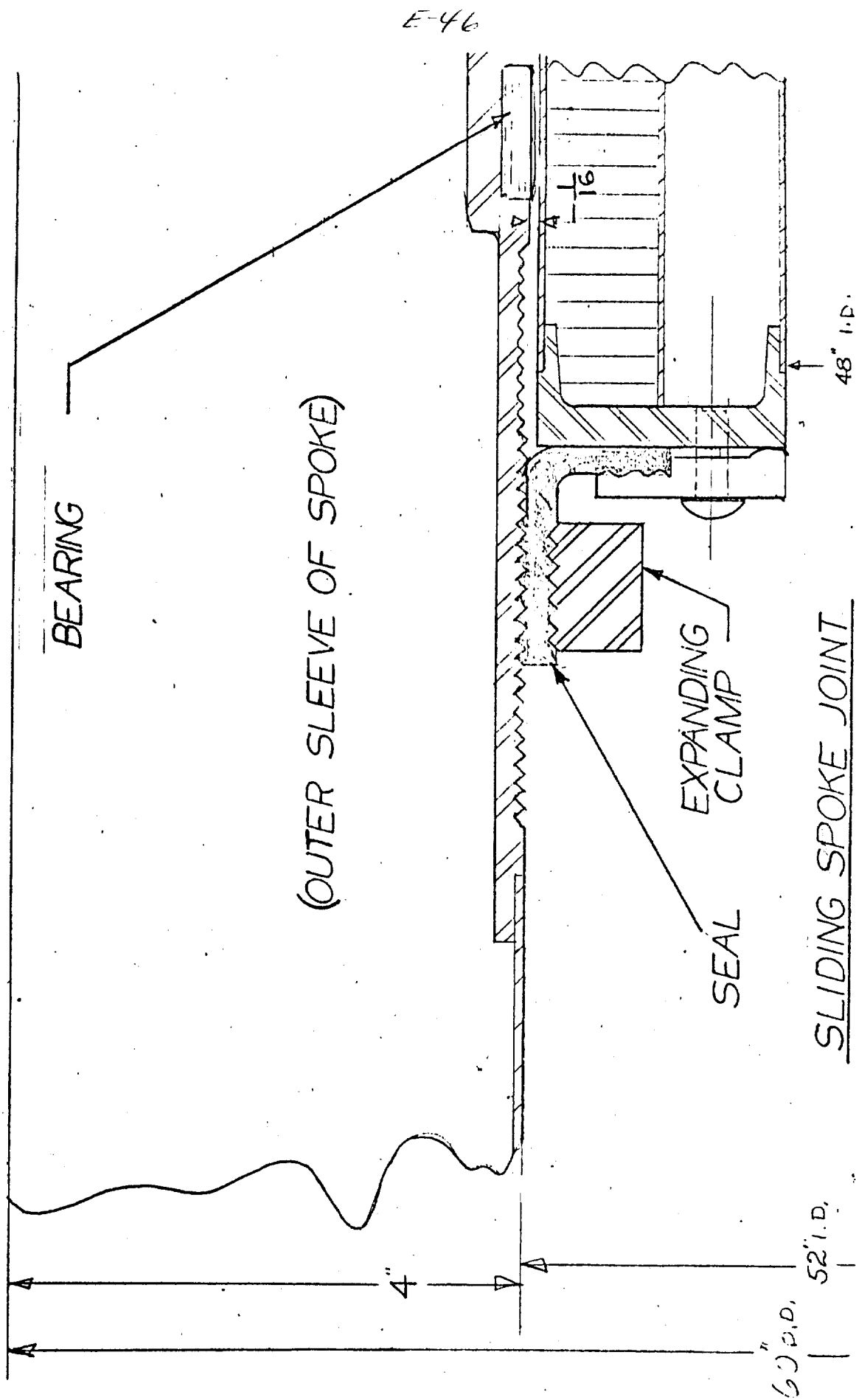
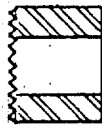
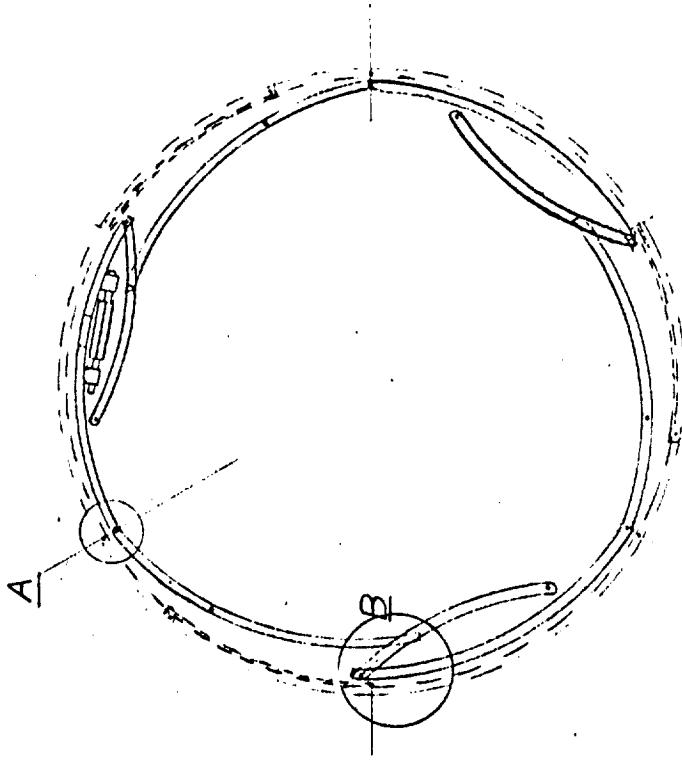
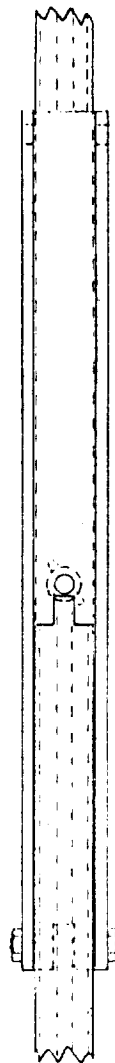
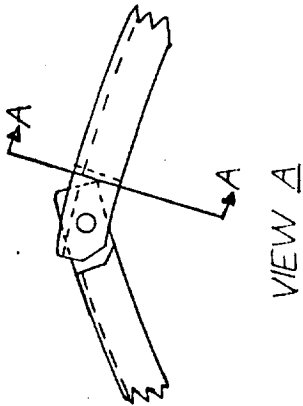


FIG. 5-16

E-47



A-A
FULL SIZE



VIEW B

FIG. 5-17
SLIDING SPOKE JOINT
EXPANDING CLAMP

ELECTRICAL AND FLUID TRANSFER CONNECTIONS

The base point space station configuration has not yet been defined to a point where the exact inventory of small lines can be enumerated.

With the present rotating and counter-rotating docking scheme it is recommended that any replenishment materials brought up by an Apollo Capsule be transferred to the station via the air lock.

It will be necessary to have transfer lines between the sections of the station carrying fluid and electric power. It is recommended that these lines be carried through the bulkheads of the pressure vessel at the air lock door joint, (Figure 6-1) so that they would be exposed to vacuum only during an emergency. The leak rate of standard fittings, as manufactured for the aircraft industry, is sufficient to permit their use.

A more difficult problem is that of carrying electric leads through the pressure vessel to solar panels, docking motors and other items outside the station. However, this area is receiving much attention and is applicable to many space vehicles. It is expected that there will be several

choices available for each type of line to be used on the space station.

Among the recommended manufacturers are:

E. B. Wiggins Oil Tool Co.

Aeroquip Company, Inc.

Hansen Company

Parker Hannifin

1. THROUGH ELECTRIC CONNECTIONS

The basic purpose of the through electric connections are twofold:

- a. To provide an electrical current pass-through in a bulkhead or door, without allowing any leakage of gas due to a pressure differential of 10 psi, in the event of section depressurization.
- b. To protect one or both of the electric connections with their own environmental protection against possible ambient conditions.

These two types of requirements, while they appear quite similar on the surface, dictate entirely different types of hardware. In case (a), for example, a simple solution could be that of running a bare piece of solid wire through a bead of insulating material that is fused into the bulk-head. Connections would be analogous to a simple

alligator clip. If the additional requirements of (b) were now applied, the resulting complexity would be extremely difficult to 'make and break'. The very simplicity of method (a) makes it desirable. The general categories of usage are twofold; power transmission and signal transmission. Both of these categories are, at present, being handled by 'Bulkhead' connectors in conventional aircraft and missiles.

The primary difference between the requirements of present usage, and those of a space station is the reliability during an extended life. What might be a satisfactory solution for a one way trip in a missile, would not be adequate for a long term orbit. Present techniques, used in manned space craft, are useful only to a limited degree and then should be scrutinized very carefully.

A conventional type of connector, at present used in bulkhead applications, would be suitable if they were sealed in such a fashion as to make them 'non-loading' or maintained in an air environment. This would mean that in these distinct areas they would be an improvement on basic bulkhead equipment.

These areas are:

- a. Seal between bulkhead and connector flange
- b. Elimination of leakage between wire and insulation
- c. Elimination of leakage through and around potting

The solution to (a) is to make the flange of the connector a permanent part of the bulkhead by means of brazing or welding.

Where consideration of rapid thermal changes negate this approach the use of an 'O' ring seal would be suitable.

In both (b) and (c) techniques have been developed for assembly and injection potting of electric connections which will solve both of these areas. In essence, high pressure injection potting, using standard electric flange couplings, and standard wire and pins, has proven satisfactory for protection in pressures up to 9000 psi, and vacuum up to 10^{-9} Torr. To utilize this technique in a vacuum it is only necessary to select a suitable injection potting compound. For purposes of rigidity and strength, it is recommended that polyethylene, or a high durometer room temperature vulcanizing rubber be used as an encapsulation material.

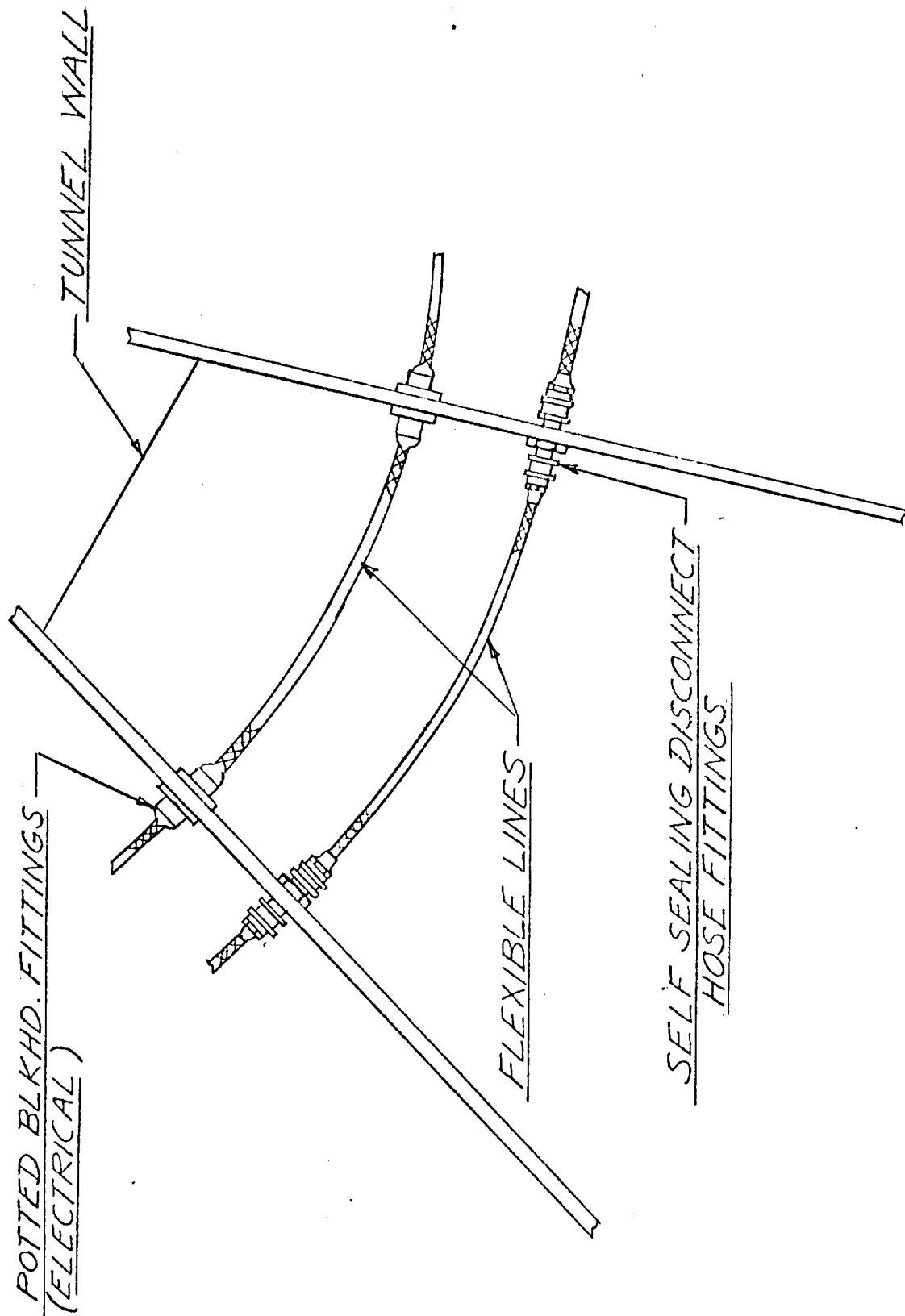
2. THROUGH FLUID CONNECTIONS

The present state of the art in aircraft hydraulic disconnects can be directly applied to the problem of through bulkhead connections in the space station. The present type of equipment for quick disconnect application will, with proper choice of elastomeric materials, be capable of supplying a make and break joint that will be satisfactory. Transmittal of fluids, both gas and liquids, will, in all probability, be specified between all modules of the space station.

While the conduit itself should be of light and durable construction, false economy insofar as weight is concerned should not be practical at points of connection. Proven and available designs and materials should be selected, furnished by experienced manufacturers.

Due care must be taken to insure that any elastomeric materials incorporated in the design will have optimal out-gassing and permeability properties.

The method of sealing the quick disconnect to a bulkhead is similar to that of the electric flange couplings. If thermal considerations permit, brazing or welding to the panel is an excellent solution.

TYPICAL FLUID & ELECTRICAL CONNECTIONS

CONCLUSIONS AND RECOMMENDATIONS

The following conclusions and recommendations have been logically evolved as a result of the study of connection and seals for a self-erecting manned space station.

1. The station one-year leak rate requirements can be met with additional experimentation necessary to provide elastomer seal leak rate versus seating stress data.
2. The seals for the space station should be of an elastomeric type.
3. Diffusion through the elastomer seals is not a determining factor since this effect is on the order of 10^2 to 10^3 less than the expected interface leakage.
4. A procedure for determining elastomer seating stress versus deflection with durometer and size as parameters has been developed for both flat gaskets and 'O' ring configurations.
5. A method for determining mean diffusion paths for elastomers 'O' rings versus deflection with durometer and size as parameters has been developed.
6. A valid method for estimating diffusion leak rate versus deflection for an elastomer 'O' ring has

been developed.

7. Polyurethane, butyl, or silicone elastomer materials should be chosen for the seals. The choice depends mainly on temperature and radiation fields.
8. An analytical determination of elastomer seal interface leak rate versus seating stress should be accomplished using as an analogy the diffusion of gas through a porous boundary.
9. Metallic seal rings cannot be used because of the high unit flange loading versus deflection and the accurate tolerance requirements.
10. Temporary sealing at the large joint interfaces should be accomplished by utilizing pressure-loaded lip seals, allowing maximum tolerance and yielding negligible flange loading.
11. The seals on the docking access airlock should be placed on the Apollo airlock section to minimize required seal life.
12. Bellows-loaded seal designs must be provided with auxiliary actuators to allow bellows compression required for subsequent seal replacement.
13. 'O' ring seal configurations should be utilized to yield optimal deflection-to-seal stress and seal stress to flange load values.

14. The 'O' ring seal cavity should have a maximum 15% voidage at operational compression to prevent extrusion and seal cutting.
15. Sliding 'O' ring seals required to make and break require a 10° to 15° tapered entrance area to minimize failure during entry.
16. Regular seal replacement should be accomplished by the astronaut crew to minimize leak rates and prevent aging-type failures.
17. Seals should be capable of being replaced without requiring station de-erection. (Reference suggested design concepts)
18. The station erection actuators and clamps should not serve the dual function of providing seal seating stress.
19. A regular procedure for connection-seal preventative maintenance should be instigated by the astronaut crew.
20. Manual clamping should be accomplished where applicable to optimize connection-seal weight and reliability. (Reference suggested design configurations)
21. The spoke sliding seal should be the expansion type to eliminate seal drag interaction during the erection sequence. (Reference suggested design configurations)
22. Manual activation of the spoke sliding seal should be accomplished for reliability and weight optimality.

23. The rotating spoke joint seal should be of dual type to allow sealing at end of rotation sequence with no functional drag during rotation. (Reference suggested design configurations)
24. The access door seal placement as depicted in the suggested design should be maintained to utilize station pressure as the seal force mode and to prevent seal degradation during astronaut and equipment passage through the doors.
25. The sliding access airlock should be incorporated in the modified Apollo replenishment vehicle. This yields increased space in the station, currently occupied by the airlock sliding section.
26. The inter-module airlock should be automatically activated and final sealing accomplished by manual clamps.
27. Pressure interlocks and gages should be provided at the access doors.
28. Simple latches are adequate for the access doors provided an initial opening stop is provided to prevent rapid door rotation due to pressure loading
29. Window nuts should be provided in the access doors to allow visual inspection.
30. Standard fluid and electrical connectors would perform adequately in the design.

31. The fluid and electrical transmission lines should terminate at the section bulkheads with quick disconnect, self-sealing connectors.
32. All access doors should be maintained in the closed condition during station operation to minimize air leakage and shock transmission during rapid depressurization.
33. Figure 7-1 following the appendices depicts the suggested connection and seal design concepts for the total station.

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DOCKING CRITERIA

These criteria are applicable for the design of docking and latching mechanisms and umbilical connections.

1. EARTH ORBIT RENDEZVOUS

- a. The rendezvous guidance and control system shall bring the target and the chaser within proximity to each other with the following tolerances at or near contact.

Axial δV + 1.5 fps
Radial δV + .5 fps
Roll, pitch and yaw rates + .05 rad/sec
Radial displacement 1.5 ft.
Roll misalignment + 5°
Axial misalignment - 5°

- 1) Each condition shall be considered separately where applicable.
- 2) Combined conditions: Each specified value plus one half the specified value for the remaining variables.

b. Strength and rigidity requirements

After docking, the joint shall be capable of supporting the following limit loads.

Bending moment - 1,791,000 in.lb.
Shear - 4,550 lb.
Axial load - +30,000, -125,000 (compress.)lb.

These loads shall be considered to act singly or in combination, whichever is more critical.

The joint shall meet the following stiffness requirements.

Effective AE
651 x 10⁶ lb. (compression)
78.8 x 10⁶ lb. (tension)

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TABLE III-1. PHYSICAL PARAMETERS OF ELASTOMER SEAL MATERIALS

Material	Coefficient of Linear Expansion $\times 10^6 \text{ in./in./}^\circ\text{F}$	Specific Heat, BTU/lb/ $^\circ\text{F}$	Thermal Conductivity BTU/hr/sq ft/ $^\circ\text{F/ft}$
Aluminum	12.7	0.22	90
Magnesium	14.3	0.25	60
Plexiglass	90	0.35	0.15
Mild Steel	6.5	0.12	27
302 Stainless	9.8	0.12	9.4
Titanium	4.0	0.14	4.3
Natural Rubber	80	0.4	0.08
SBR	100	0.4	0.14
Buna N	70	0.4	0.14
Neoprene	70	0.4	0.11
Butyl	90	0.4	0.05
Silicone	150	---	0.12
Fluorocarbon	80	---	---
Urethane	100	---	---

TABLE III-2 ELASTOMER SPECIFICATIONS

<u>Specification</u>	<u>Class</u>	<u>Suitable or Required High Polymer</u>	<u>Remarks</u>
MIL-R-1149A		Neoprene SBR Butyl Buna N	Rubber Sheets, Strips, and Gaskets; Solid, Synthetic, Medium and Medium Hard
MIL-P-5315A		Buna N	Packing, O-ring, Hydro- carbon, Fuel Resistant
MIL-G-5510A		Buna N	Gasket; Straight Thread Tube Fitting, Boss
MIL-P-5516B		Buna N	Packings Gaskets, Preformed, Petroleum Hydraulic Fluid Resistant
MIL-R-5847C	I	Silicone	Extreme Low Temperature Resistant
	II	Silicone	High Temperature Resistant
MIL-R-6855	I	Buna N	Fuel Resistant
	II	Neoprene	Oil Resistant
	III	SBR	Non-Oil Resistant
	IV	Buna N	Oil Resistant (For Contact with Acrylic Plastics)
	V	SBR	Non-Oil Resistant (For Contact with Acrylic Plastics)
MIL-R-7362B	Comp. A, Low Temp.	Buna N	Synthetic Oil Resistant
	Comp. B High Temp.	Buna N	Synthetic Oil Resistant

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<u>Specification</u>	<u>Class</u>	<u>Suitable or Required High Polymer</u>	<u>Remarks</u>
MIL-P-25732		Buna N	Packing, Preformed, Petroleum Hydraulic Fluid Resistant, 275°
MIL-R-25897		Viton	Rubber, High Temper- ature Fluid-Resistant
MIL-R-25988		LS-53	Rubber, Silicone, Oil and Fuel Resistant

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TABLE III-3 GAS PERMEABILITY CONSTANTS $\times 10^7 \frac{\text{cm}^2}{\text{sec.atm.}}$
AT ROOM TEMPERATURE

High Polymer	N ₂	O ₂	Air	H ₂	CO ₂	He	CH ₄
Natural Rubber	2	2	1	3	3	3	2
SBR	0.5	1	0.7	3	9	2	2
Butyl	0.08	0.05	0.1	0.4	0.2	0.9	---
Hypalon	---	---	0.7	---	---	---	---
Neoprene	0.1	0.3	0.1	1	2	0.6	0.2
Buna N	0.02	0.07	0.1	0.2	0.4	0.1	0.04
Thiokol FA	---	0.02	---	0.03	---	---	---
Silicone	70	---	10-70	---	200	100	---
Poly FBA	---	---	2	---	---	---	---
Urethane	0.04	0.1	0.05	---	---	---	---
Vyram	---	---	0.007	---	---	---	---
Polyethylene	0.09	0.33	---	0.8	---	0.6	---
Teflon	0.02	0.8	---	2	---	60	---

TABLE III-4 ELASTOMER PHYSICAL PROPERTIES

<u>Material</u>	<u>Durometer Hardness + 5 Shore 'A' Degrees</u>						
<u>Natural Rubber</u>	30	40	50	60	70	80	90
Ultimate Strength Tensile (psi)	2,200	3,500	3,700	4,400	4,100	3,300	
Ultimate Elong. %	725	680	640	620	490	310	
Compression Set ASTMB 158°F	40	8	10	12	15	26	
<u>Neoprene</u>							
Ultimate Strength Tensile (psi)		2,500	3,500	2,800	3,700	3,000	
Ultimate Elong. %		750	620	350	250	250	
Compression Set		25	15	10	10	5	
<u>NBR (Buna N)</u>							
Ultimate Strength Tensile (psi)	1,700	1,900	2,100	2,100	2,000	2,100	2,200
Ultimate Elong. %	860	630	750	570	530	120	80
Compression Set	87	85	79	74	71	36	33
<u>Butyl</u>							
Ultimate Strength Tensile (psi)		1,800	2,200	2,000	2,300	2,100	
Ultimate Elong. %		800	750	700	600	500	
Compression Set		15	20	20	30	30	
<u>Urethanes</u>							
Ultimate Strength Tensile (psi)			2,000	3,000	4,000	5,000	6,000
Ultimate Elong. %			650	500	450	450	500
Compression Set			40	9	12	15	20

BELLOWS DESIGN CRITERIA

A bellows can be used to provide the required seal face loading and are supplemented by springs in some applications.

Metal bellows can be designed to make the seal face contact pressure a function of the internal pressure by locating the radial position of the sealing faces with respect to the effective diameter of the bellows. End-face seating force is influenced by the pressure effect from the internal pressure if the bellows mean diameter is larger than the diameter of the sealing face.

A bellows can be considered analogous to a spring mass. Critical frequencies are a function of the bellows mass and spring rate. Avoid using a metal bellows seal having a natural frequency below or equal to the frequency of vibration of the system.

SPRING RATE If the spring force of the bellows is used to hold the end faces of the seal together, it is necessary to calculate the effective spring rate, K_e , of the bellows. The bellows free length can be specified to be some length, L_f , greater than the operating dimension. Equivalent spring rate, K_e #/in. for a flat disc bellows can be estimated from

$$K_e = \frac{\pi E D}{N_d} \left(\frac{t}{W} \right)^3$$

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E = modulus of elasticity (psi)

D = mean bellows diameter (in.)

N_d = number of discs

W = disc width (in.)

For bellows with V-shaped convolutions

$$K_e = \frac{4ED}{3h^{0.5} W^{2.5} N_d}$$

h = half corrugation pitch (in.)

Pressure effects:

S_t = Total stress range (psi)

$$S_t = \frac{3Et\delta}{W^2 N_d} + \frac{PW^2}{2t^2}$$

P = maximum internal pressure (psi)

δ = total bellows movement from free length (in.)

Fatigue life (N) = no. cycles to failure.

$$N = \left(\frac{1.6 \times 10^6}{S_t} \right)^{3.5}$$

Table IV-1 lists the design parameters for various commonly used bellows materials.

PROPERTIES OF BELLOWS MATERIALS

TABLE IV-1

	Stainless Steel(321)A	Stainless Steel(347A)	A286	AM350 HT	Inconel X A-Aged	17-7PH TH 1050	Titanium	6061-T6
Density								
#/in-3	0.29	0.29	0.288	0.28	0.30	0.276	0.16	0.098
Linear -320-68	7.5	7.5	8.3	6.6	5.7	5.6	3.85	10
Coefficient 32-212	9.3	9.3	9.37	6.9	7.6	6.3	5.7	13
of Expansion 32-600	9.5	9.5	9.67		8.4			14.1
in/in/of 32-1200		10.6						
X10 ⁶								
Yield Strength (PSI) X10 ⁻³								
70° F	35	40	89.3	170	92	181	120	40
-300° F	42	54	135	52(A)	119	40(A)		47
1200° F	22	22	62.5	217	82	216	63(1000° F)	7(500° F)
				140(700° F)		160		
Ultimate Strength (PSI) X10 ⁻³								
70° F	90	95	105.1	203	162	196	130	45
-320° F	200	195	200	156(A)	192	130(A)	235	60
1200° F	50	50	103.5	248	120	237	78(1000° F)	8(500° F)
				210(700° F)		230		
Modulus of Elasticity X10 ⁻⁶ (PSI)								
28	27.6(-320° F)	27.6(-320° F)	29.1	30.4(80° F)	31(80° F)	29(70° F)	18.96(-320° F)	10.2(-320° F)
	25 (70° F)	25 (70° F)		25.4(600° F)	23.1000° F)		16.5 (75° F)	9.8(212° F)
	22.5(1000° F)	22.5(1000° F)					10.1 (100° F)	8.0(500° F)

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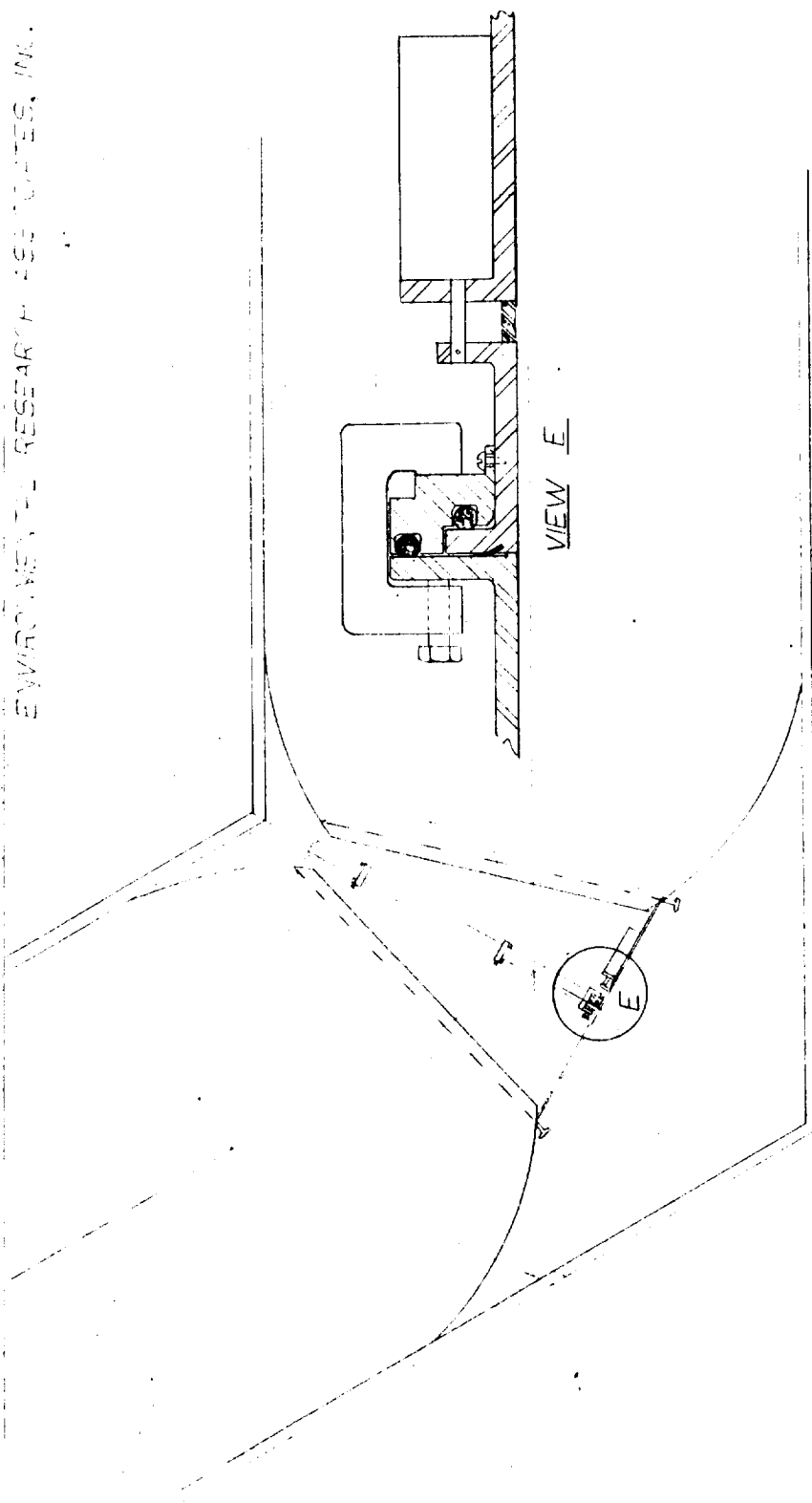
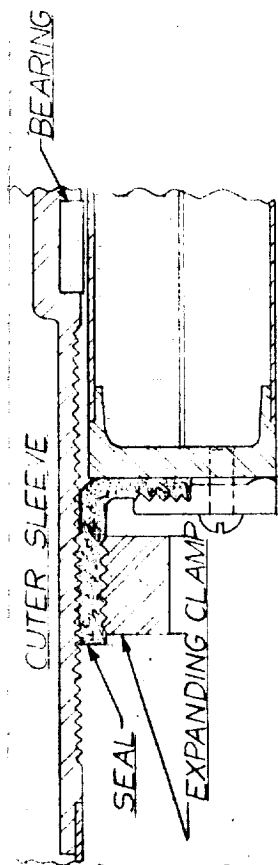


FIG. 7-1
OVERALL CONNECTION SEAL
DESIGN CONFIGURATION

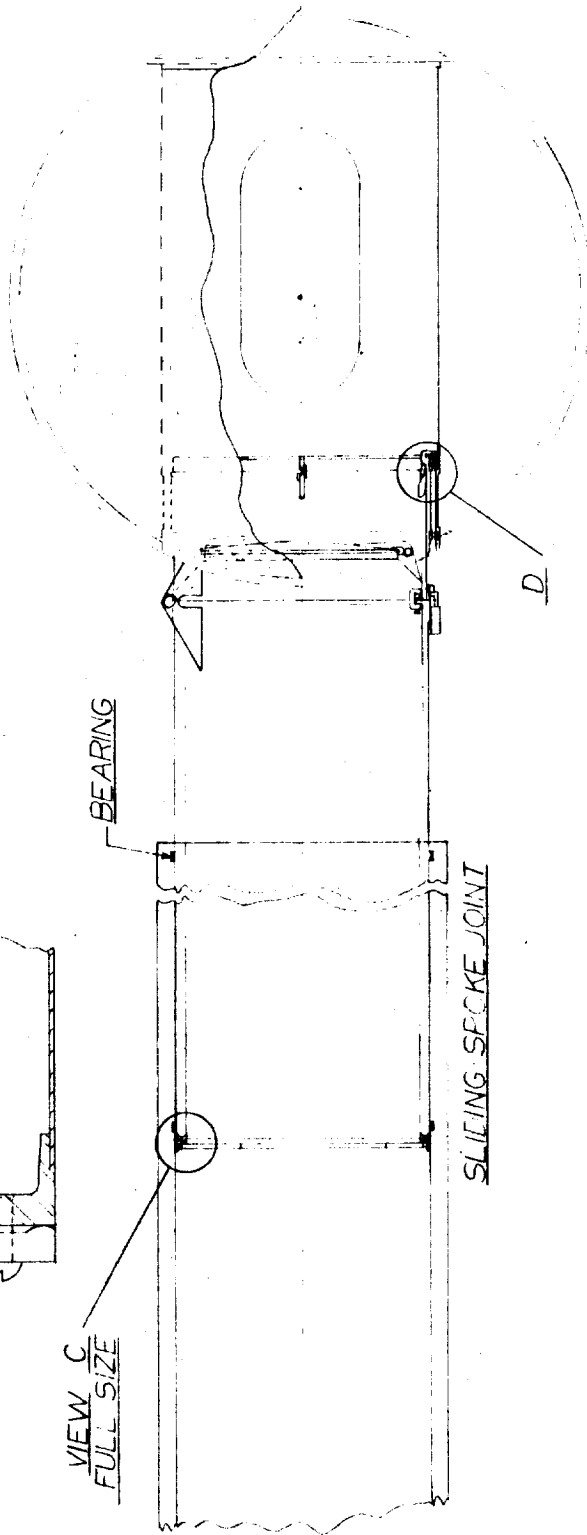
INTERMODULE AIRLOCK
1/2C



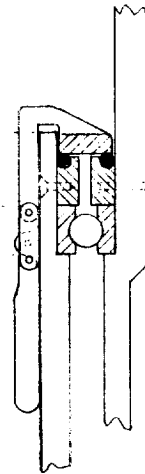
SPOKE TO MODULE HINGE JOINT

VIEW C
FULL SIZE

BEARING



ROTATING JOINT

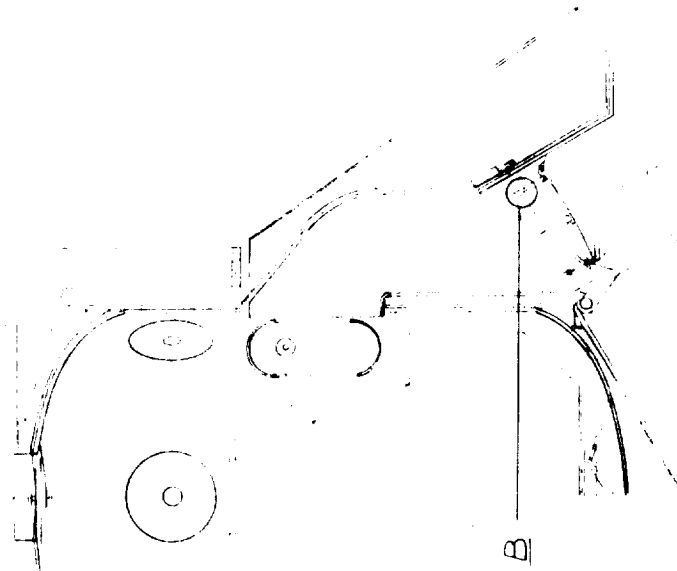
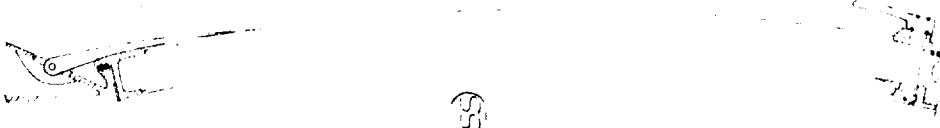


VIEW D
1/3 SIZE

EXTENDED SPOKE TO MODULE
1/20 SIZE

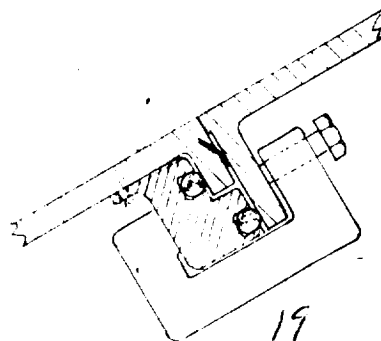
SECRET

(TOP SECRET) (S)



SPOKE TO HUB HINGE JOINT
140 SIZE

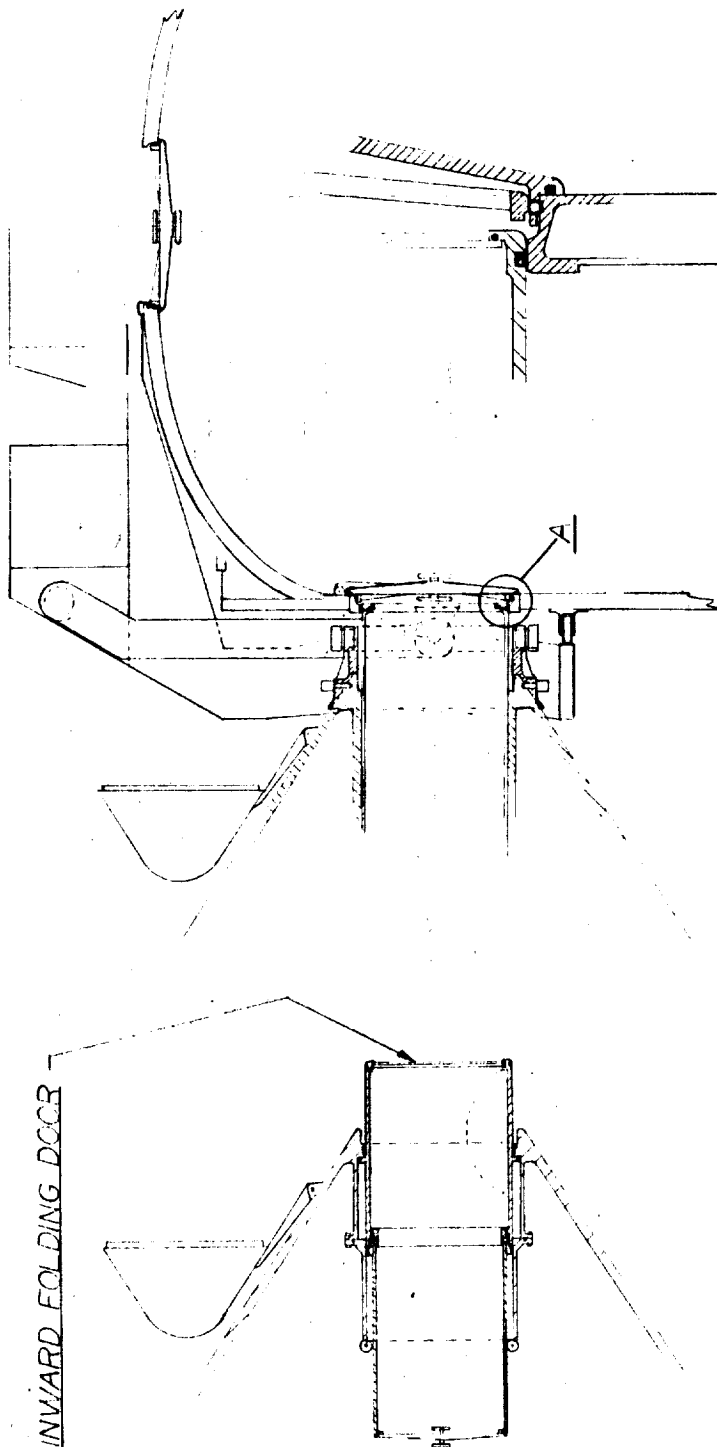
1.40 SIZE



VIEW B
1/2 SIZE

1/2 SIZE

INWARD FOLDING DOOR



DOCKING ACCESS AIRLOCK
EXTENDING PASSAGEWAY

1/20

APOLLO VEHICLE STOWAGE

1/20 SIZE

VIEW A
1/5 SIZE

*Environmental
Research
Associates*

